POWER

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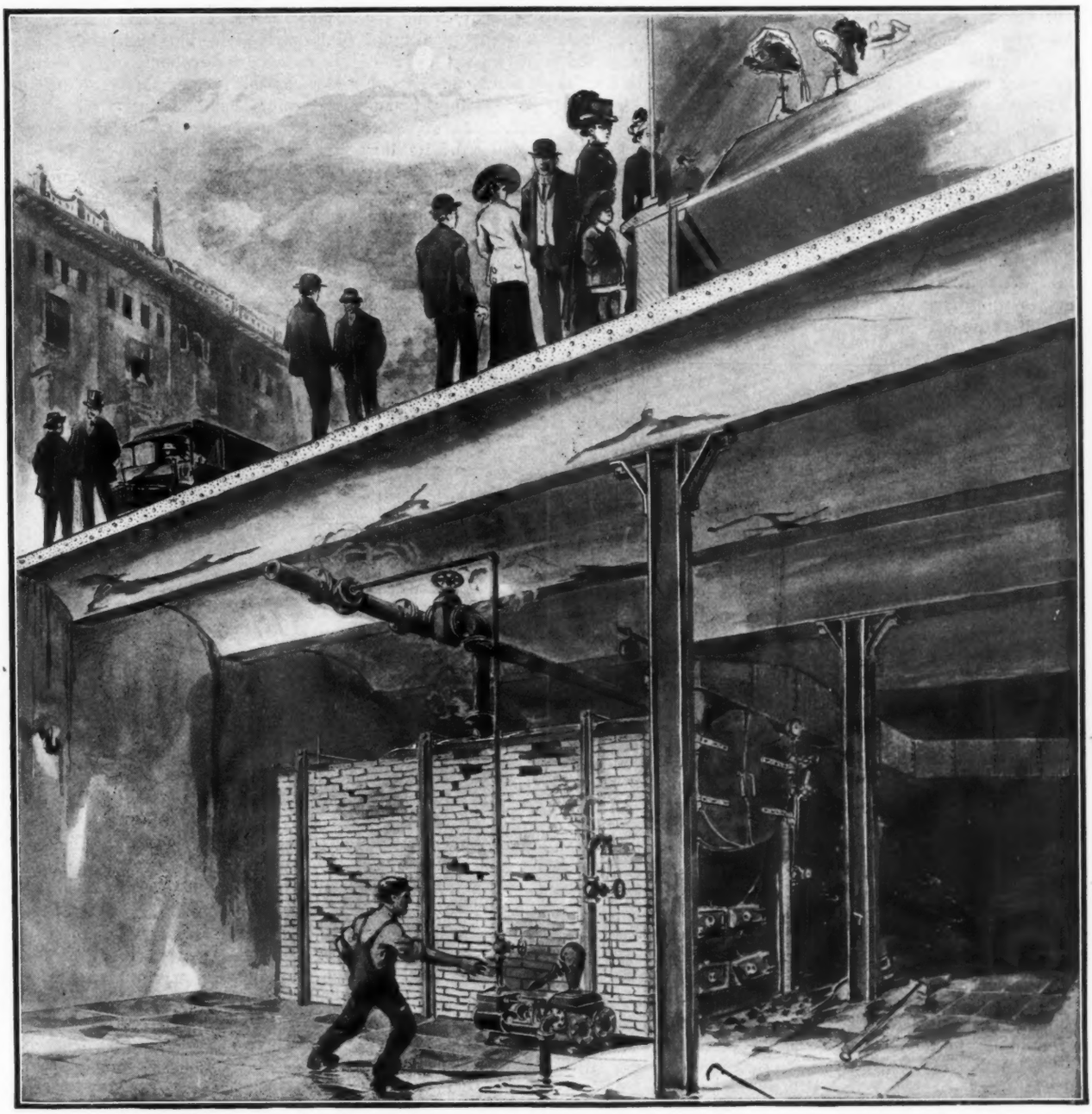

How an unsuspecting public is menaced by dangerous lap-joint, return tubular boilers located under sidewalks. Doubly dangerous are the boilers when operated by incompetents.

Lentz System Applied to Steam Engine*

By Sigfried Rosenzweig

The Lentz engine which is now used extensively in Europe, employs valves of the poppet type both for admission and exhaust. These permit the employment of high superheat as a result of which some exceptionally low steam consumption rates have been obtained.

Thermodynamics teaches that the efficiency of an engine increases when the temperature limits between which it works are increased. That occurs in a steam engine when steam enters the cylinder at the highest temperature and leaves it at the lowest possible. Hence, to obtain economy in a steam engine three factors are of greatest importance: First, high steam pressure; second, high superheat, and third, high vacuum.

The greater the steam pressure in a steam engine, the greater is its capacity to develop power. By increasing the steam pressure from 100 to 200 lb., the output of the engine can be practically doubled, or it can do the same work as before with much less steam. Steam at 200 lb. pressure contains only 10.8 B.t.u. per lb. more than steam at 100 lb.; so if the coal has 14,000 B.t.u. per lb., only about $^1/_{1000}$ part of a pound of coal is necessary to raise the steam pressure from 100 to 200 lb. As the high-pressure boiler costs very little more than the low-pressure boiler, these facts forcibly point toward the adoption of high steam pressures. Modern stationary engines use boiler pressures from 175 to 200 lb.

That the use of superheated steam raises the efficiency of an engine is generally acknowledged. The reasons for it are various, chief among which is the reduction of initial condensation and the leakage past valves and piston. As these two items represent from 25 to 50 per cent. of the total steam consumed in an engine, their reduction is important.

When dry saturated steam enters the cylinder, it has to come in contact with surfaces which have just been in contact with cold exhaust steam and which are covered with a film of moisture. As water serves as a ready means of transferring heat from the steam to the metal a great quantity of the incoming steam is wasted by condensation. The use of superheated steam reduces this initial condensation, extensive experiments having shown that superheated steam, because of its low density, is a poor heat conductor and that quite naturally this property becomes more and more marked the higher the superheat.

Professor Doerfel, of the University of Prague, found that by increasing the superheat from 46 to 256 deg. F. the loss during the admission period decreased from 8400 to 1900 B.t.u. Higher superheat will probably, apart from radiation, prevent almost entirely the loss of heat due to the exchange of heat between cylinder walls and steam. These experiments were confirmed by another authority, Seemann, who found that by superheating from 0 to 360 deg. F. the heat given up to the cylinder walls decreased from 36.2 to 9.3 per cent. Another experiment showed that with a superheat of about 210 deg. F. it made no difference as to the initial condensation whether the engine was running condensing or noncondensing, though in the former case the temperature drop due to the expansion of steam amounted to 195 deg. against 135 deg. F. in the latter.

These experiments, which were confirmed by many other authorities, show conclusively that it is advisable to use high superheat to obtain high economy. The author from his own experience has found that a superheat of from 200 to 250 deg. F. is quite commercial and gives very satisfactory working results. The higher initial cost for installing superheaters is easily counterbalanced by the smaller piping, as superheated steam allows of considerably higher steam velocities, by the greatly reduced heating surface in the boilers due to the vastly superior economy in the steam engine, by the greater safety of the power plant, due to the absence of water in the pipes and engine, and by the greatly reduced coal consumption.

To illustrate this point may be mentioned the power plants of the electricity stations located in Vienna and in Rome. The former has twenty piston-valve engines of 1000 hp. each. When the superheaters were installed, the high-pressure cylinders had to be reconstructed and were fitted with Lentz valves and gear. The outlay of about $80,000 was recovered within one year from the saving of coal. In the power station at Rome the installation of Lentz engines and superheaters increased the output of the plant by about 30 per cent. The introduction of superheat further simplified greatly the design of steam engines, so that compound engines with highly superheated steam successfully replaced triple and quadruple expansion engines, thus producing a saving in initial cost, of floor space and running expenses.

The application of superheat has been restricted for various reasons, chiefly because of the bad effect of the high temperature on valves unsuitable for superheat, but these obstacles were never sufficient to blind engineers as to the merits

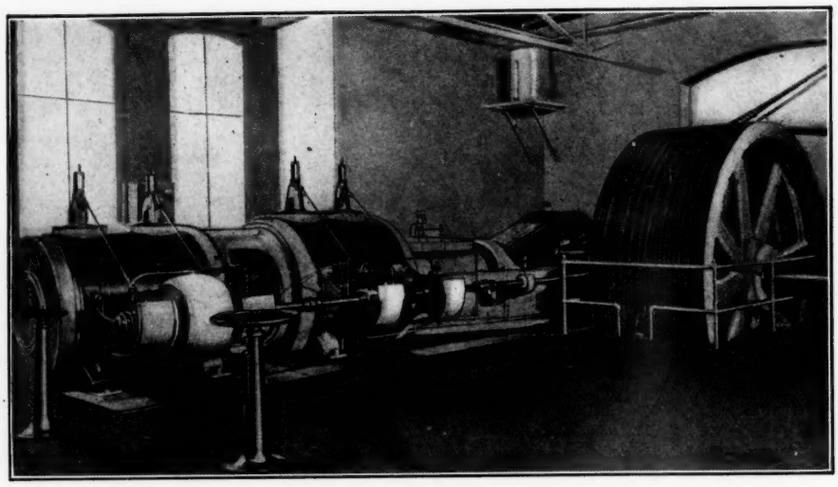

FIG. 1. VALVE-GEAR SIDE OF LENTZ ENGINE

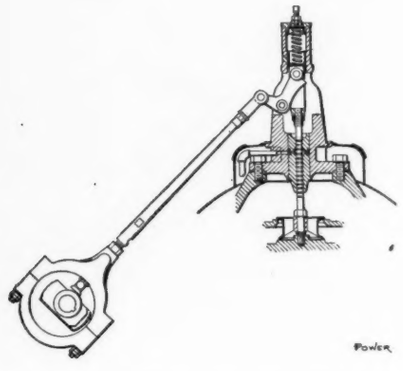

FIG. 2. SECTION THROUGH VALVE

*From a paper read before the Ohio Society of Mechanical, Electrical and Steam Engineers.

of superheated steam. Owing to the expansion of the metal all valves which have to rely upon the surrounding walls for steam tightness are unsuitable for superheat. For high steam pressures and high degrees of superheat a balanced and frictionless valve has been used, and the deficiencies of the slide, piston and Corliss valves can only be remedied by the adoption of the balanced multiple-seated poppet valve.

The advantages of the poppet valve are no rubbing surfaces, hence no lubrication and its adaptability for high temperature; small weight and a practical balance, hence its adaptability for high speed and high steam pressure; no possible wear, consequently the economy of the engine is maintained for an indefinite period.

Probably the best representative of this type of engine is the Lentz, a view of which, from the valve-gear side, is shown in Fig. 1. A layshaft runs alongside the engine, driven from the main shaft by a pair of bevel gears. All the eccentrics are keyed on this layshaft, each separate eccentric operating one valve by means of an eccentric strap rod and lever. In this way the different valves are entirely independent from each other and can be set very accurately according to the requirements of steam distribution.

Fig. 2 shows the different parts of the internal valve-gear, the valve, the spindle with its grooves, and spindle guide with roller and cam lever. All the parts, as pins, bushings, rollers and cams are case-hardened to prevent wear. The valve is entirely balanced but for the width of the seat, which even in the larger sizes does not exceed ⅛ in.. and hence takes very little power to operate. The valve-gear consists only of a roller and cam lever. The latter is oscillated by means of the eccentric rod and is so curved that it opens the valve and closes it very rapidly. No dashpots are used as the whole operation is positive. On account of the noiseless operation of the valve-gear and its great simplicity it is possible to operate the Lentz engine safely at high speeds, a limit to which has not been reached at even 350 r.p.m. A feature is the absence of any spindle packing, as the spindles are simply ground into cast-iron bushings and provided with grooves, thus producing a labyrinth system in which the condensed steam and oil collects and by which the escape of any steam is successfully prevented. A similar principle is used for the piston-rod packing, which is made entirely of cast-iron rings but so arranged as to follow the floating action of the piston rod.

A section through the cylinders of a tandem compound engine is shown in Fig. 3. The low-pressure cylinder is placed next to the bed. This arrangement gives the high-pressure cylinder a chance to expand freely without being obstructed by the distance piece and the low-pressure cylinder. The opening in the distance piece is made large enough to remove the back cover and piston of the low-pressure cylinder without disturbing the alignment of the engine.

All the low-pressure and high-pressure exhaust valves are driven from fixed eccentrics, the high-pressure steam eccentrics only being influenced by the governor.

Fig. 2 shows also the arrangement of the high-pressure steam eccentric, which is arranged to slide on a block keyed to the shaft. When the load changes, the governor slides the eccentric on the block, and alters the angular advance and the throw of the eccentric according to the requirements of the load, while the linear lead remains constant. The governor is placed between the two steam eccentrics and influences their position without the help of a belt or chains.

The governor is shown assembled in Fig. 4. It consists of two pendulums pivoted to a support which is keyed on the shaft. By means of links the pendulums are connected to the governor inertia body, which can move freely around the shaft, being connected with the shaft only by a flat circular swing which counterbalances the centrifugal forces of the pendulums. The potential energy stored up in the inertia body is converted to kinetic energy as soon as the slightest change of load and consequently of speed occurs. As the inertia forces and the centrifugal forces act in the same sense their combination produces a governor of a very sensitive and instantaneous action. It is possible to change the speed of the engine while in motion by means

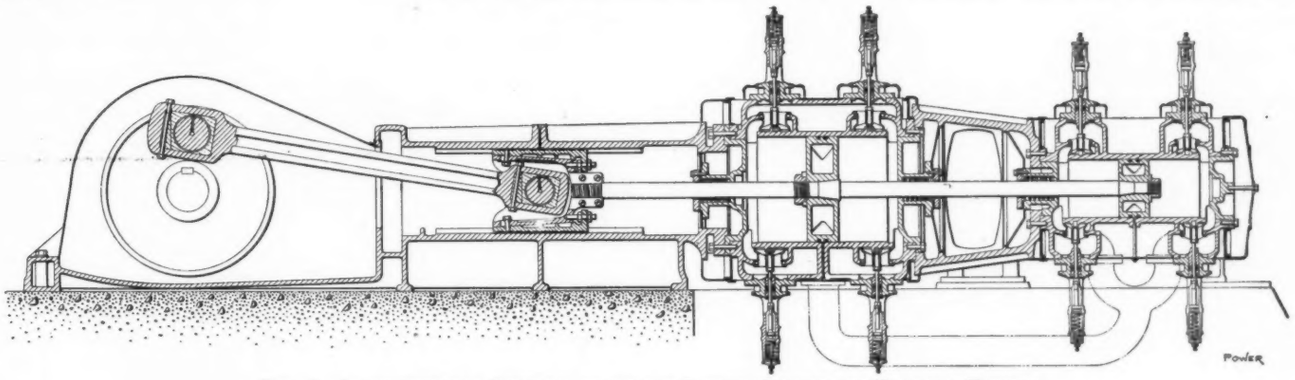

FIG. 3. LONGITUDINAL SECTION THROUGH LENTZ COMPOUND TANDEM ENGINE

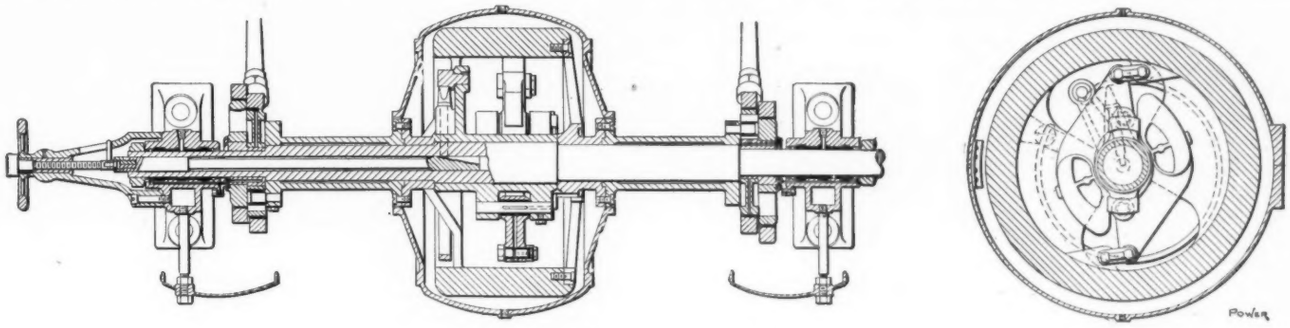

FIG. 4. LENTZ GOVERNOR

of a radial pin which presses against the spring and changes its tension. The radial pin can be moved by means of a shaft and handwheel which is placed at the end of the layshaft.

Extensive tests have been carried out on a Lentz simple engine having a 15-in. cylinder and 21-in. stroke. This engine showed a steam consumption of 19.2 lb. per i.hp.-hr. with an average steam pressure of 145 lb. gage, an average superheat of 47 deg. F. and a back pressure of 2 lb. This performance gives a Rankine efficiency of practically 76 per cent.

A tandem engine carefully tested gave the following results: The first test with saturated steam, 170 lb. boiler pressure, and 26 in. vacuum, gave 366 i.hp. with a steam consumption of 12.3 lb. per i.hp.-hr. The second test with the same conditions as before but with superheated steam of 150 deg F. showed a steam consumption of 10.4 lb. per i.hp.-hr. These results were obtained from an entirely new engine within two days of first running.

The Lentz system has also been extensively used in connection with semi-stationary engines. a combination of an engine with boiler, superheater, condenser, air pump, feed-water heater and feed pump, in other words, a compact, high-class and self-contained power plant which is extensively used in isolated plants in Europe and South America. On account of its compact arrangement and the absence of any piping and losses incurred by it, the economy of these engines is remarkable. One of these engines was tested by Prof. Grassmann, of the Technical College of Carlsruhe, Germany. The 200-hp. unit gave a steam consumption of 7.4 lb. and a coal consumption of 0.92 lb. per b.hp.-hr., figures which probably stand as world records. The mechanical efficiency of the engine was 93 per cent. and the combined efficiency of boiler and superheater 82 per cent.

Power Plant Records

By G. E. Miles

The articles published from time to time have clearly shown that the best way for an engineer in an isolated plant to keep the central station from getting his business is, first, to operate his plant at its highest efficiency and, second, to be able to show his employers what the power is costing them.

It has also been shown that daily records are necessary to enable the engineer to accomplish the above results, and with the hope that some may be helped, the following suggestions and sample daily report are submitted. The fact that this form was designed for, and used by, a small central station should deter no one from adapting it to the use of an isolated plant. Often the best way to fight fire is with fire, and certainly the isolated plants can learn much from the central stations that will enable them to better compete with the latter.

At the top of this form are spaces for each engineer to record the time of his going on and off watch. The first column of blank spaces is for the record of the load on the plant. If recording wattmeters are used of a form that can be read closely every half hour, the load can be best determined from them, as the output for the period is desired rather than the instantaneous load at the time of reading. If there are no wattmeters from which the load can be determined, the load can be estimated from the ammeter readings.

The second column is for the steam pressure and the third and fourth show which engine was running. The latter is conveniently done by drawing a line from the time the engine is started until it is stopped. The fifth, sixth and seventh columns are for the arc-lamp circuits. Isolated plants will rarely need these columns for arc lamps, but they may need them for other purposes.

From the data in the first column, curves can be plotted showing graphically the load on the plant at any time. These curves, although not essential, are nevertheless very convenient.

At the top on the right-hand side the weight of coal used is recorded. Many engineers will balk at this because they have no means of weighing their coal, but means can be provided at a small expense and these data are absolutely essential.

At the plant where this form was used small platform scales were adapted to the purpose by building a platform for the scales large enough for a wheelbarrow to be run upon it. As the coal had to be wheeled into this plant, the coal was weighed with little extra trouble. At another plant an ordinary washtub was provided with a bail and hung on a steelyard that would weigh 150 lb. of coal at a time. When small coal is used it can be measured in a tub with sufficient accuracy if a tubfull is frequently weighed to keep a check on the weight. Weighing the coal and keeping the records will cause the engineer additional labor, but the satisfaction of knowing

DAILY STATION REPORT

ENGINEERS

NAME	FROM	TO

DATE ..... 19....

A.M.	K.W.	Steam	Eng. No.1	Eng. No.2	Cir. No.1	Cir. No.2	RR	P.M.	K.W.	Steam	Eng. No.1	Eng. No.2	Cir. No.1	Cir. No.2	RR
12								12							
30								30							
1								1							
30								30							
2								2							
30								30							
3								3							
30								30							
4								4							
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5								5							
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9								9							
30								30							
10								10							
30								30							
11								11							
30								30							
12								12							

COAL

12 to 6 .....
6 to 6 .....
6 to 12 .....
Total .....

WATT METER READING

Midnight .....
6 a. m. .....
Noon .....
6 p. m. .....
Midnight .....

WATER METER READING

Noon .....
6 p.m. .....
12 midnight .....
6 a.m. .....

Boil. No. ..... On..... Off.....

Cylinder Oil, ... Qts
Engine Oil, ... Qts
Waste, ... Lbs
Packing, ... Lbs

Boilers Cleaned .....
Repairs, ...

REPORT INTERRUPTION OF SERVICE ON OTHER SIDE

..... CHIEF ENGINEER

what results he is securing in the operation of his plant will amply repay an ambitious engineer.

The next spaces are for the record of the wattmeter readings at six-hour periods. If no wattmeters are provided, the output for these periods can be computed from the readings recorded in the first column, provided the load is not subject to frequent and large changes. If there are such changes, a recording wattmeter is necessary to determine the output with reasonable accuracy.

Where boiler-feed water is paid for on a meter basis, the readings of the water meter should be recorded in spaces provided for that purpose.

In the absence of wattmeters the load on a direct-current plant may be easily found by multiplying the current in amperes by the voltage and dividing by 1000 to reduce to kilowatts. The load on an alternating-current plant cannot be as easily or as accurately determined without wattmeters as the power factor is seldom known, and when three-phase generators are used the phases are seldom balanced.

The following approximation will serve to give an engineer a comparison from day to day, and will be much better than no information as to what the plant is doing: Multiply the amperes by the primary voltage and the product by the power factor for single-phase circuits; the power factor will be between 75 and 95 per cent. For three-phase circuits take one-third of the sum of the amperes on three phases and multiply by the volts and this sum by 1.732, also multiply by the power factor. Care should be taken to get the correct primary voltage, which is the voltage as shown by the meter multiplied by the ratio of the transformer supplying the meter. This ratio is usually 20, so that when the meter reads 110 volts the primary volts are 2200, but if the meter reads 115 volts the primary volts are 2300.

An engineer may make the measurements mentioned and record the data without deriving much benefit from doing so, but if he studies the figures and computes the cost of power in pounds of coal per kilowatt-hour and then compares the results at different times he will be able to see what conditions give the best results and also to keep a check on the work of his subordinates.

All this means more work for the men, who probably feel that they are doing more than they are being paid for, but it will pay in the end. It is the man who knows what he is doing, and can do, who can confidently ask his employer for the pay he knows he is earning.

The Horsepower of a Pipe Line

By R. S. Bayard

The relation between static and friction head, flow and velocity. Determining the proper velocity and flow to get the greatest horsepower out of a line of given size with a given static head.

"How much water will flow through a 2-in. pipe under 100 lb. pressure?"

Many readers will recognize this as a familiar question, which they themselves have asked or have been called upon to answer. The trouble is that the question is not complete. Upon inquiring where the pressure is measured, the usual reply is: "Oh, right at the end, or perhaps 6 in. back from the end." With water flowing out of an ordinary open pipe without any end obstruction it is absolutely impossible to obtain a pressure of 100 lb., or any other pressure above atmospheric, at the end, or even 6 in. back.

If the water is flowing out rapidly its energy of velocity will exert a pressure in the direction of its motion, as in driving a Pelton waterwheel; but, without a contracted nozzle, the pressure against the walls of the pipe at its end must be atmospheric.

To produce a flow, there must be pressure somewhere. Then, where should this pressure be measured, and what becomes of it at the end of the pipe?

If a pipe wall offered no frictional resistance to the movement of the water, no pressure at all would be required to make it flow along the pipe to its open end, and in fact, it would be impossible to build up any pressure even with a pump, no matter how much water was being handled, nor how long the pipe.

What necessitates pressure, or "head," to cause a flow through an open-ended pipe is friction between the water and the pipe walls. This friction depends upon the length and diameter of the pipe, and the velocity of the water through it. The greater the length, the more the friction, and so with the velocity. Increase in pipe diameter, other things remaining unchanged, decreases the friction.

So, before answering the original question, it must be known how long is the pipe and at what point the pressure is measured. As an example, consider what might be a Western mountain hydraulic power plant. Imagine a pipe line down the side of a mountain and let the lower end of the pipe be 450 ft. below the dam which catches the water and directs it into the pipe. Also let the length of the riveted steel pipe be 6000 ft., or a little over a mile, and its diameter 16 in. Such a combination of conditions is not out of the ordinary.

Assume that the pipe is of uniform diameter throughout its length, and for the present has no waterwheel nozzle at its lower end, but is provided with a gate valve giving when open practically no obstruction to the flowing water.

First, imagine that the gate valve at the lower end is closed, that there is no flow at all and that the pipe is full to its upper end, 450 ft. higher than the gate valve. Under these conditions the pressure at the gate valve is equivalent to the "head" of water above it. It is

$$450 \times 0.433 = 195 \textit{ lb. per sq.in.}$$

Pressures in such a pipe line are usually expressed in head of water, or feet of vertical water column. Thus, with the gate valve at the lower end closed, the head of water is 450 ft. Now, assume that the valve is partially opened, allowing a slight flow of water. The reading of the pressure gage on the upstream side of the valve will scarcely be affected, but if the valve is opened wider, allowing an appreciable flow, the gage reading will decrease. The reason is that the flowing water produces a friction force on the walls of the pipe, which diminishes the previous gage pressure. Part of the 450-ft. head is being expended in "friction head.'

If the valve is opened still wider, the flow will become greater, increasing the velocity of the water along the pipe walls and increasing the friction. The increased friction head will further decrease the original static head, so that the gage reading will be still lower.

Up to this time the partially opened valve has exerted appreciable resistance to the flow of the water, this becoming less the greater the flow, because the friction head helps to hold the water back. As the valve opening is increased the gage reading will continue to decrease until finally the valve will be opened wide and the gage will register no pressure.

Under these conditions the flow of water will be so great that all of the static head will be absorbed in friction, with the exception of a very few feet, which is represented by the increased velocity of the moving mass of water. At this time it may be truly said that the quantity of water is all that will flow through the given pipe with the original given pressure, and this quantity would be the answer to such a question as that originally asked.

To get a visual idea of these condi-

tions, the accompanying curves have been plotted. In this diagram, horizontal distances represent cubic feet of water flowing per minute through a 16-in. pipe, 6000 ft. long, with a total static head of 450 ft., while one of the vertical scales so marked represents feet head of water pressure. For the curve marked "Head Lost in Friction," the left-hand scale gives the feet of head absorbed in friction and the corresponding quantity of water flowing may be read from the lower scale.

It may be observed that for a flow of 200 cu.ft. per min., the friction head is about 12 ft., which subtracted from 450 ft. leaves 438 ft. available head, as shown by the curve marked "Available Head at Lower End." When the flow is 500 cu.ft. per min., the friction head is about 65 ft., leaving an available head of 385 ft. It is this available head that exerts the pressure on the gage at the lower end of the pipe. Its reading under the last condition would be about

$$385 \times 0.433 = 167 \text{ lb. per sq.in.}$$

This means that the friction is absorbing 28 lb., and the remaining 167 lb. is being resisted by the partially open valve at the exit. At this rate of flow the velocity is so low that its energy represents a negligible part of the head.

The curve further shows that with the gate fully open, and offering no resistance at all to the flow, the head lost in friction will reach the full value of the original static head of 450 ft. All the original head has now been used up in forcing the water at high speed along the pipe whose friction holds it back. The gage will show no pressure at the lower end of the pipe. This takes place when the flow is practically 1400 cu.ft. per min.

When the water is flowing at the above rate its velocity through the 16-in. pipe will be

$$V = \frac{Q}{A};\ V = \frac{1400 \times 144}{201 \times 60} = 16.72 \text{ ft. per sec.}$$

the area of a 16-in circle being 201 sq.in. At this speed the velocity head, or portion of the static head used to impart this much energy to the moving column of water, is

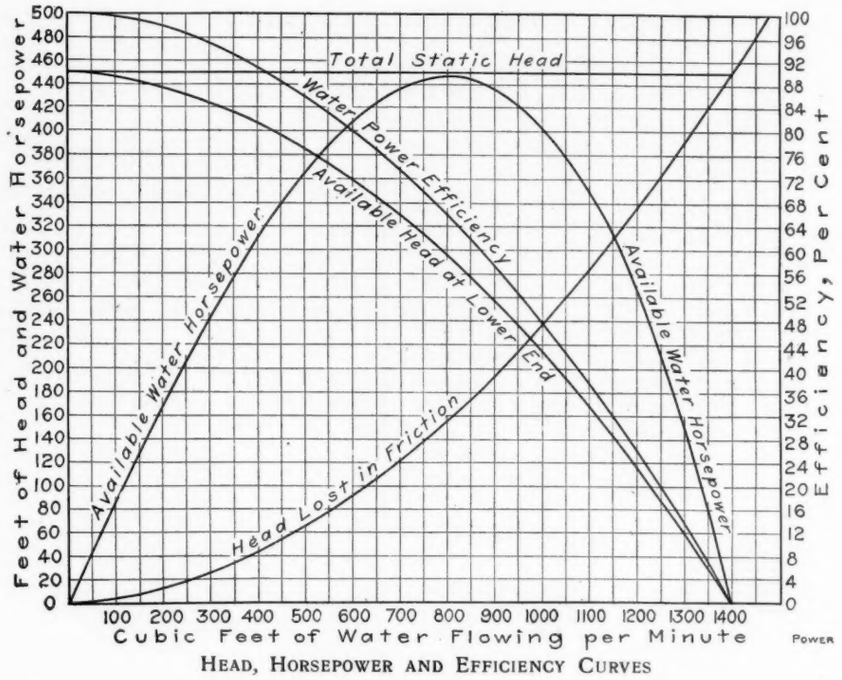
$$h = \frac{V^2}{2g} = \frac{16.72^2}{2 \times 32.16} = 4.35 \text{ ft.}$$

As this quantity is so small compared to the total head of 450 ft. and as the actual coefficient of friction must be guessed at for a new installation, in the absence of experiments under the actual existing conditions, this quantity may safely be neglected. This is especially so for working conditions, as less than half this quantity would be allowed to flow to a waterwheel, because to do any work from its original head of 450 ft. some of this head must be left available at the pipe exit or wheel. With half the quantity, there would be half the velocity and so one-fourth the velocity head, or not much over 1 ft. This is far within the "guessing limit" of the actual friction head.

The available head at the pipe exit brings up the question of what velocity to use and what flow to allow to get the greatest horsepower out of a pipe line of a given size, with a given static head. To figure available power of the flowing water, it is necessary to multiply the pounds of water flowing per unit of time by the head available at the bottom end of the pipe after the friction head has been deducted.

As an example, with the 6000 ft. of 16-in. pipe with 450 ft. of static head, let the flow be 500 cu.ft. per min. The curve gives a friction loss of 65 ft. and so an available head of 385 ft. The water horsepower then would be

$$W.hp. = \frac{500 \times 62.4 \times 385}{33{,}000} = 364$$



HEAD, HORSEPOWER AND EFFICIENCY CURVES

If the flow is allowed to be 800 cu.ft. per min., the lost head of friction will be about 155 ft. and the available head 295 ft. The horsepower will be about 446. A waterwheel would not be able to deliver all of this, due to its own imperfections and so this available power would be further reduced by the mechanical efficiency of the wheel. The further reduction would amount to somewhere around 20 per cent., with a first-class wheel.

It will be observed that there are two conditions of flow where there would not be any horsepower available at all. These occur when there is no flow, with all the head available, and when there is the maximum flow, with all the head used up in friction.

Under this last condition there would be some power due to the velocity head already spoken of, or the energy of the moving mass. At the maximum flow of 1400 cu.ft. per min., the velocity head was found to be equivalent to 4.35 ft., so the power would amount to about

$$\frac{1400 \times 64.4 \times 4.35}{33{,}000} = 11.9 \text{ hp.}$$

This is a very small percentage and as a matter of fact, it is doubtful if this pipe could ever deliver 1400 cu.ft. per min., because the curves show the entire 450 ft. of static head used up in friction at this capacity; whereas part of the available head, after deducting for friction, must be represented in the velocity of the flowing mass. At 1390 cu.ft. per min. the velocity would be 16.6 ft. per sec., and the velocity head would be

$$\frac{16.6^2}{2 \times 32.16} = 4.28 \text{ ft.}$$

At this point also the friction loss is just about this much short of the total 450 ft. of static head, so all of the static head would be represented in friction and velocity. However, all of these close figures are "hair splitting" on such work and the two theoretical points of no horsepower available are about as the curves show.

What, then, is the best velocity to employ in such a pipe line to develop power, or what is the maximum power available with a given size and length of pipe, and at what water velocity will this be obtained, assuming that the necessary volume of water is available?

To answer this question, a glance at the chart shows that the maximum horsepower is about 447 and it occurs when the friction head amounts to about 155 ft. This loss by friction is just a little over one-third of the total head available. It can be mathematically shown that the maximum horsepower will be

available in a given pipe line, when about one-third of the static head is sacrificed to friction.

Such a condition may be said to give the maximum efficiency of the pipe, though it certainly is not the maximum efficiency of the use of the water. A larger pipe would, of course, give the same flow with less friction loss, or a greater flow with the same loss, but it also would be true of the larger pipe that it would deliver its maximum power when the friction loss through itself became one-third the total head. It is only a matter of how much money it is desired to spend on the pipe line, and whether or not it is more economical to pay for a bigger pipe, used below the point of its maximum horsepower and thus use the available head to better advantage.

By assuming that there is no friction loss and the water power is being used at 100 per cent. efficiency, and so on down to zero efficiency with all the head lost in friction, a curve such as that marked "Water Power Efficiency" in the chart is obtained.

From the efficiency curve, it may be noticed that the water-power efficiency is about 66 per cent. at the point of maximum water horsepower. This naturally follows from the fact that one-third of the static head is sacrificed to friction. As previously stated, there will be the further loss of efficiency of the waterwheel, or turbine, amounting to 20 per cent. or more, giving an overall efficiency of somewhere around

$66 \times 80 = 5280 = 52.8$ *per cent.*

Of course, if the money is available, and a higher efficiency is wanted, a pipe may be used that will reduce the friction loss to, say 10 per cent., giving an overall efficiency in the seventies. These matters are all subjects for study to suit results to costs, and each case requires individual analysis and judgment by the designer.

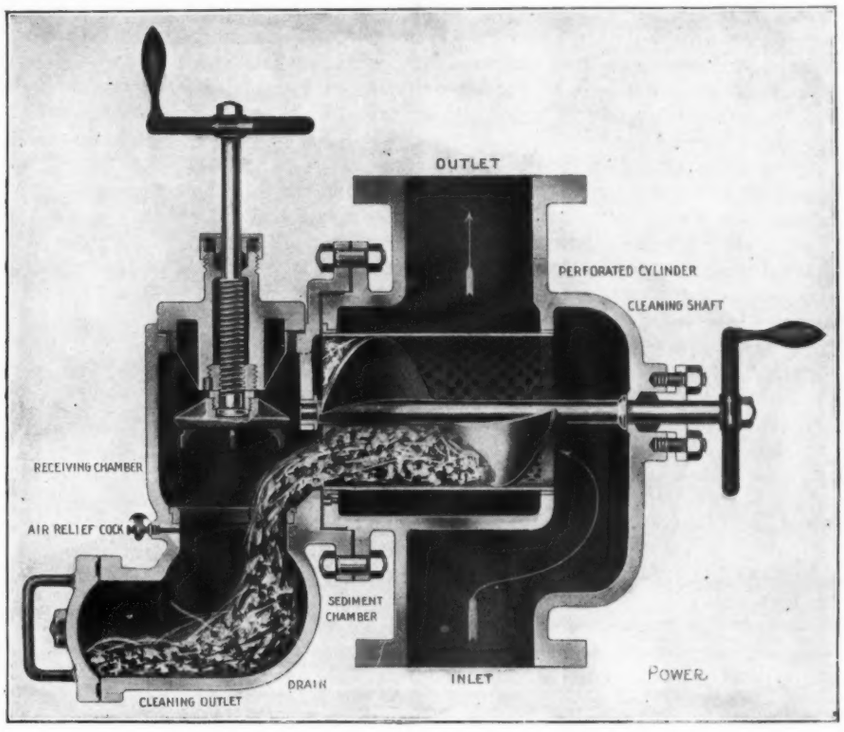
Holt Automatic Strainer

This device is designed to automatically remove all kinds of solid foreign matter from water supply lines without interrupting the flow of water. It consists of a cast-iron casing, having an inlet and outlet opening and a perforated brass cylinder, having open ends, interposed between the inlet and outlet of the casing. The liquid to be strained flows into one end of the perforated cylinder and passes out through the perforations into the outlet of the casing.

Foreign matter is retained on the inside of the perforated cylinder until the cleaning shaft is rotated. The blade of the cleaning shaft, which has a $\frac{1}{32}$-in. clearance between it and the inside wall of the perforated cylinder, sweeps the inner surface and forces the foreign matter out of the opposite end from that through which the liquid enters the receiving chamber, from where it falls by gravity into the sediment chamber.

There is only one valve, and this need be operated only occasionally, when the sediment chamber is to be cleaned out. In practice, all that is required to keep the strainer cleaned, is to occasionally turn the handwheel connected to the cleaning shaft about two complete revolutions, and whenever necessary, clean out the sediment chamber, both operations being performed without interrupting the flow of water through the strainer.

This strainer is made by the Rosedale Foundry & Machine Co., North Side, Pittsburgh, Penn.



SECTION THROUGH AUTOMATIC STRAINER

Water Tank Support Collapses

On the morning of Nov. 12, a 45,000-gal. wooden tank collapsed at Pile Brothers Starch Works, Indianapolis, Ind., putting the plant out of commission. In falling it completely wrecked the pump house, and four double-acting water pumps were badly damaged and will require extensive repairing before they can be used again. Fortunately nobody was injured. The falling tank also broke a 4-in. steam line and three flanges on a 10-in. steam line, making it necessary to shut off steam on all buildings. Water from the tank ran into the engine room and flooded the wheel pits.

Four columns supported the tank, each made in three sections of 9-in. iron pipe, ¼ in. thick and 20 ft. long. The joints were made with couplings, ⅝ in. thick and 9¾ in. long. Diagonal and horizontal braces ran from opposite corner couplings. The horizontal braces were made of 4-in. pipe, ⅛ in. thick, and the diagonal braces were of 1 in. round iron, threaded at the ends, and screwed into the cast-iron fittings. Each rod had a turn-buckle in the center.

There were indications that one or more of the diagonal braces were subjected to a stress greater than their elastic limit, which allowed one or more of the columns to buckle, breaking one of the couplings near the top and leaving the tank unsupported on one side. The damage is estimated at $5000.

The first mention of the occurrence of coal in the United States, according to the United States Geological Survey, is made in the journal of Father Hennepin, a French Jesuit missionary, who in 1679 recorded the site of a "cole mine" on the Illinois River, near the present city of Ottawa, Ill. The first actual mining of coal was in the Richmond Basin, Va., about 70 years after Father Hennepin's discovery in Illinois, but the first records of production from the Virginia mines were for the year 1822, when, according to one authority, 54,000 tons were mined.

Reports of the anthracite coal trade are usually begun with the year 1820, when 365 long tons were shipped to Philadelphia from the Lehigh region. Prior to this, however, in 1814, a shipment of 22 tons was made from Carbondale, also to Philadelphia. It is probable that the actual production prior to 1820 was between 2500 and 3000 tons.

The production for 1911 was 496,221,168 short tons.

Progress of Wolf Locomobile Engines

By F. E. Junge

In the last fifty years steam pressures have increased from eighty or ninety pounds to over two hundred pounds. The average horsepower of the locomobiles manufactured at the Wolf shops has increased from 8 hp. to 80 hp. They turned out 90,000 hp. of locomobiles last year.

The firm of R. Wolf, Magdeburg-Buckau, Germany, has just published a memorial, edited by the well known historian Conrad Matschoss, which gives a comprehensive survey over the field of its various activities during the last fifty years. The following is a short abstract of the development as it relates to the "Locomobile," or integral steam plant in which the engine boiler and accessories are mounted together in a concrete unit.

There are at present 88 types of locomobile engines built by the firm, differing in detail in so far as they are destined for use either as stationary or portable machines, simple or compound, for use with superheated or saturated steam, condensing or noncondensing, and, of course, varying in capacity within the range of these various applications. Though elasticity of production is an indispensable condition of German manufacture, which has to satisfy the exacting requirements of extensive markets at home and abroad, there is a growing tendency toward standardization, an attempt to make the various parts for the different types interchangeable, in order to reduce the total number of parts used, and thereby the cost of manufacture, to a minimum.

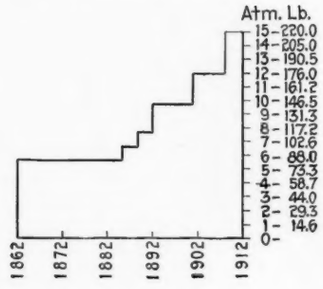


FIG. 1. INCREASE IN BOILER PRESSURES

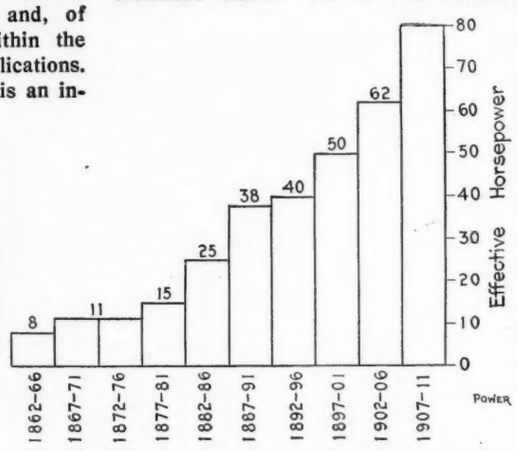


FIG. 3. GROWTH OF AVERAGE HORSEPOWER OF LOCOMOBILES

Fig. 1 shows the gradual increase of boiler pressures employed in the various periods of 50 years. It is seen that during the first 20 years the boiler pressure remained at 6 atmospheres (88 lb.) while today a pressure of 15 atmospheres (220 lb.) is employed in some types. The indicator diagrams assembled in Fig. 2 will give a better idea of the evolution of the locomobile and of the improvements realized than would a survey of the details of construction. As the diagrams are all drawn to the same scale a comparative analysis is possible. Of the denominations used, n means the number of revolutions per minute, p is the boiler pressure, p_i the mean indicated pressure and t the temperature of the steam when entering the high-pressure cylinder.

In this table diagrams 1 to 4 refer to single-cylinder locomobile engines using saturated steam. No. 1 was obtained from an engine having flat slide valves and throttle regulation. These engines have been built since 1862, the card being taken in 1878. The weaker lines in the diagram represent lower loads. Diagram 2 is from an engine having a Meyers hand-adjustable cutoff valve and throttle regulation. The type has been built since 1874, the card was taken in 1878. Diagram 3 refers to an automatic cutoff engine with Rider cutoff valve and Porter governor. The card was taken in 1880. No. 4, taken in 1892, refers to the same system of governing as the foregoing, but to a boiler pressure of 10 atmospheres (147 lb.).

Nos. 5 and 6 refer to single-cylinder locomobile engines using superheated steam. The distribution of the steam is in all types effected by means of a piston valve with elastic rings, the travel of this valve and its angle of advance being adjusted by a governor acting directly on the driving eccentric. The first engines of this type were built in 1898. The diagrams numbered 6 were taken at various loads.

Diagrams 7 to 10 refer to compound locomobiles using saturated steam. Nos. 7 and 8 were obtained from engines using Rider valves on the high-pressure and flat slide valves on the low-pressure side. Nos. 9 and 10 were taken from engines built since 1892, having Rider valves on the high-pressure and Trick canal slide valves in the low-pressure cylinder.

Diagrams Nos. 11 to 14 refer to compound locomobiles using superheated steam. They are all equipped with piston valves controlled by the governor. Nos. 11 and 13 refer to engines having flat slide valves and 12 to one having Trick's slide valve in the low-pressure cylinder. No. 14 is from the latest type of engine in which the exhaust from the high-pressure cylinder as well as the admission into the low-pressure cylinder are controlled by the same piston valve.

In the 50 years which have passed since the inauguration of the firm the maximum output per unit of locomobile engines manufactured has increased from 20 hp. in the period between 1862 and 1866 to 900 hp. in the period between 1907 and 1911. If the total capacity of all locomobiles turned out in the same

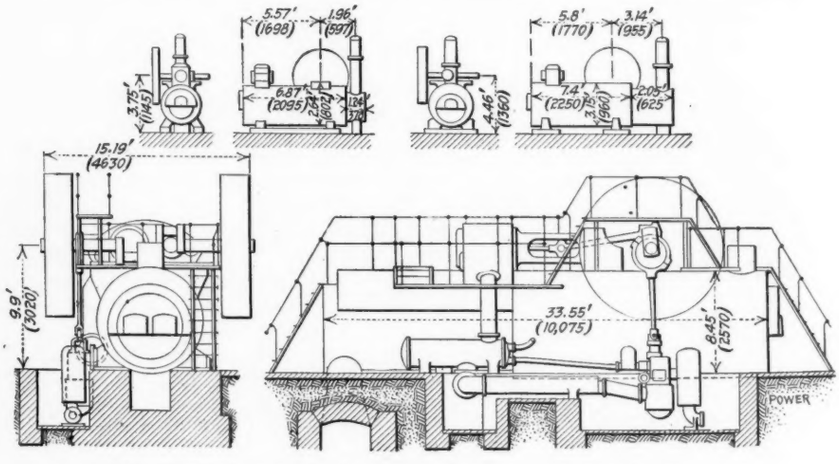


FIG. 5. SMALLEST AND LARGEST STATIONARY LOCOMOBILE

period of time is divided by the total number delivered, the average capacity of the locomobiles manufactured is obtained. Fig. 3 shows how remarkable has been this growth of average output; from 8 hp. in the first five years to 80 hp. in the last five years. The total capacity of all locomobiles turned out by the Wolf firm in 1911 was 90,000 hp.

Though, owing to its peculiar combination of boiler and engine in one unit, the locomobile type of engine is limited in its power output, this remarkable growth of capacity per unit has been effected without changing the principal design and construction of the engine. Fig. 4 gives a graphic representation of this development, showing the smallest portable locomobile ever built as well as the smallest and the largest standard type in stock. Fig. 5 shows the corresponding development of stationary locomobile engines.

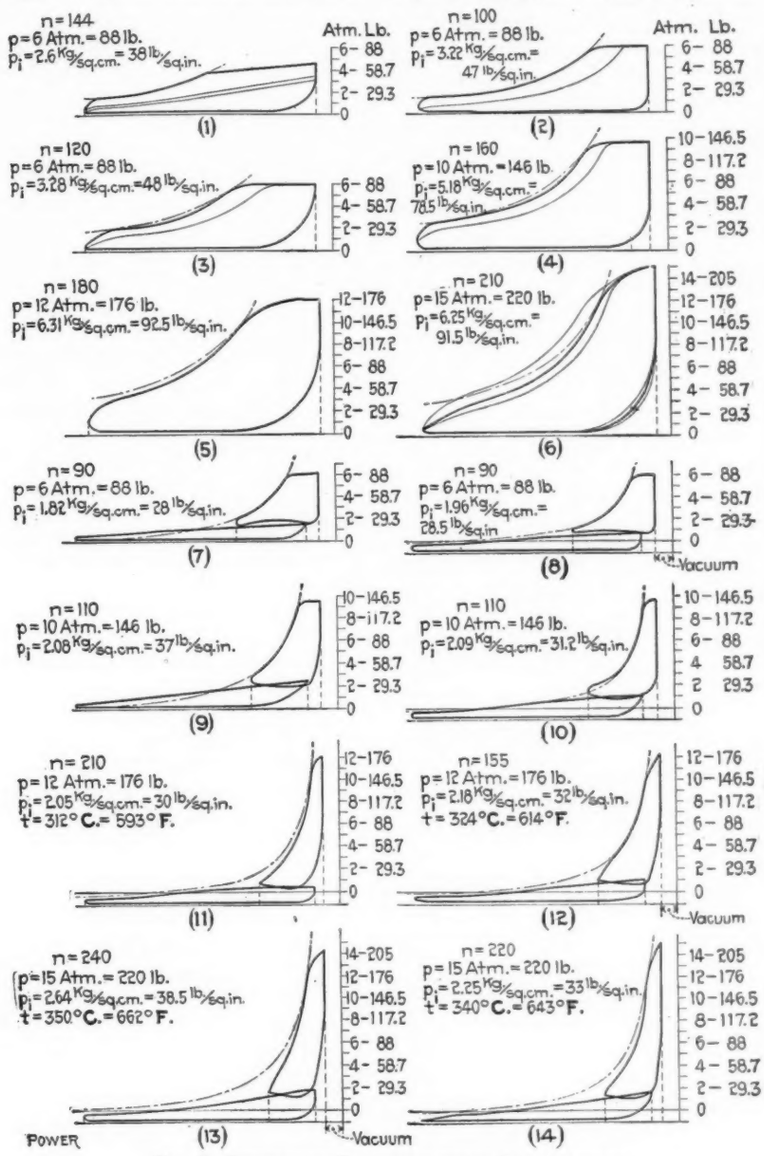


FIG. 2. INDICATOR DIAGRAMS FROM LOCOMOBILES

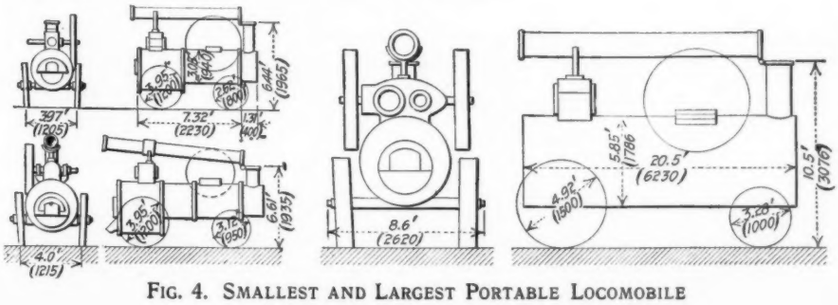


FIG. 4. SMALLEST AND LARGEST PORTABLE LOCOMOBILE

Considerable attention has been devoted to the reduction of fuel consumption, one of the most important factors of operating cost. How successful has been this attempt is shown in Fig. 6, giving the results of tests made by eminent experts in the course of 30 years. Fifty years ago the consumption of coal per horsepower-hour was hardly less than 5 kg. (11 lb). Today guarantees of not to exceed 0.5 kg. (1.1 lb.) for the larger types; or one-tenth the amount of the original consumption may be had. With unusually high degrees of superheat consumptions as low as 0.358 kg. (0.79 lb.) per hp.-hr. are obtained with 100-hp. units.

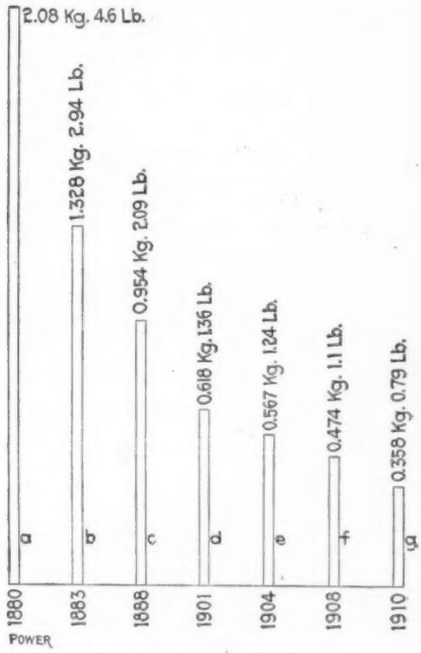


FIG. 6. REDUCTION IN COAL CONSUMPTION

It is interesting to compute the saving thus realized in pecuniary equivalents. With coal prices such as obtain in Germany the annual saving in fuel cost thus realized by improvements in the course of 20 years for a 50-hp. locomobile amounts to 3031 marks or $722. The total saving of coal so made possible represents a considerable conservation of the natural wealth.

This survey over a field of industry in which American manufacture has not even made a feeble attempt to compete, cannot be concluded without reflecting, in the briefest possible manner, upon the causes which exist and are responsible for the lack of initiative in this line of endeavor. For one who has been concerned with the comparative study and analysis of technical progress in Germany and in the United States for the last ten years, it seems unfortunate that, owing to the abundance of fuel resources there is no urgency from within, and, owing to the restrictive tariff of 45 per

cent. on manufactured goods, there is no urgency from without, compelling manufacturers to concentrate their energies in the direction of efficient service to consumers.

It is safe to say that the stagnation which is making itself so unpleasantly felt today in various lines of mechanical engineering in the United States, among others in the realm of high-grade power-plant apparatus, is due primarily to this lack of influx and stimulus from abroad which, when admitted would force many a manufacturing concern to abandon its old-fashioned cut-and-try methods and move on, if it wanted to be sure of a competency and a profitable business.

Unless these conditions change, American consumers are certain to receive inferior service for their money, and American manufacturers are sure to be left behind in the conquest of foreign markets. Because, as long as there is an alternative to unsatisfactory dealing, foreign consumers will ask for the best things money can buy, not for the things which it is most profitable for manufacturers to sell. If they can choose between an inefficient standard engine and an efficient modern machine they are sure to invest their money where it is likely to yield them the greatest immediate returns. The present system of seclusion reduces to an inferior status both the American producer and the American consumer and is bound to work infinite harm to the country, unless by a reasonable change of commercial policy, it is amended—the sooner the better.

Setting Corliss Engine Valves

By J. K. McIntyre

Directions for setting the valves of a Corliss engine for the first time, according to the method employed by erecting men.

The valve-gear adjustments are illustrated by drawings.

Most engineers know how to adjust the valves of a Corliss engine already in service, but the method of setting the valves and making adjustments for the first time on a new engine is perhaps hazy to all but a few of the most experienced engineers.

In the following the shop method of setting the valves of a single-eccentric noncondensing engine is described.

Wristplate Movement

First, adjust the eccentric rod for length, so that the rocker-arm travels an equal distance each side of the central position; see Fig. 1. Then turn the eccentric on the shaft until the rocker-arm is plumb, set the wristplate central and clamp it in position. Adjust the length of the hook rod to suit, and scribe a line on the hub of the wristplate and another to correspond on the wristplate stand, the latter mark being made light so that it can be changed later.

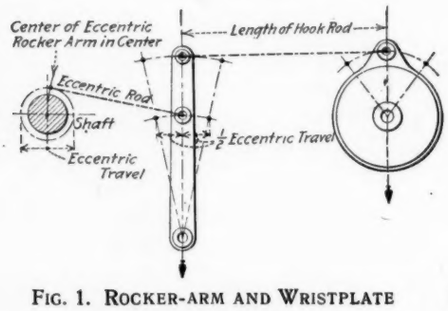


Fig. 1. Rocker-arm and Wristplate in Central Position

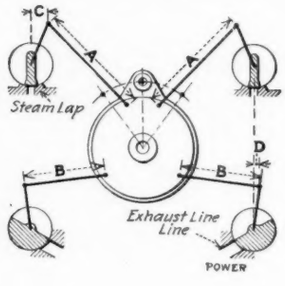


Fig. 2. Steam and Exhaust Valve Adjustment

Throw the eccentric ahead to the outermost travel and mark the wristplate stand in line with the mark on the wristplate hub, to show the extreme travel on one side of the center; turn the eccentric to the opposite extreme and scribe a mark on the wristplate stand to show the opposite extreme travel. This applies only to new engines. Next adjust the length of the steam and exhaust connections *A* and *B*, to suit *C* and *D*, Fig. 2. By using the dimensions *C* and *D* rather than following the working drawing for the length *A* and *B* any variation in the location of the ports is overcome.

Setting for Lap

With the wristplate plumb, set the steam and exhaust valves to the dimensions given on the working drawing and then mark the valve stems for the keyways, cut and drive the keys into place.

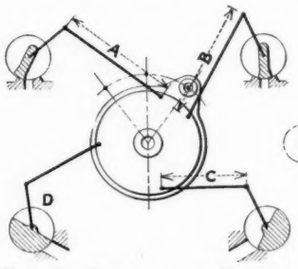


Fig. 3. Steam and Exhaust-valve Adjustment. Full Port Opening

While setting the steam valves, it is important to see that the releasing gear latch is locked up with the steam arm as when the valve is in operation.

When resetting the valves of an old engine, in which the keyways have been cut in the valve stems, and also to determine the steam "lap" as nearly as can be done without specific figures, proceed as follows: Set the wristplate plumb and adjust the exhaust rods *B*, Fig. 2, so that the exhaust-valve edges are line and line with the port edges; turn the wristplate to its extreme right-hand travel and adjust the length of the steam-valve rod *A*, Fig. 3, bringing the edge of the valve to a fullport opening. Turn the wristplate to the opposite extreme travel and repeat for valve *B*. Then center the wristplate and note if the steam laps at each valve are the same; if not, equalize them.

Adjusting the Dashpot Rod

Next, for either new or old engines, turn the wristplate to the right-hand extreme travel and adjust the dashpot rod for the valve *B*, bringing the hardened-steel block of the releasing-gear hooks to a hooked-on position with $\frac{1}{32}$-in. clearance; turn the wristplate to the opposite extreme travel and adjust the dashpot rod for the valve *A*.

Turn the crank shaft until the crank is nearly on the head-end dead center.

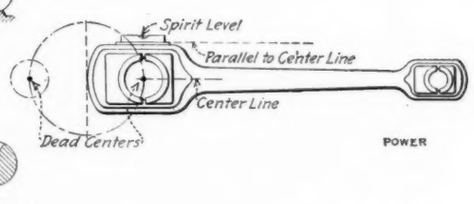


Fig. 4. Leveling the Crank

Lay a spirit level on that part of the connecting-rod which is parallel with the center line of the rod, Fig. 4, and turn

the crank shaft until an exact level is indicated. Then turn the eccentric in the direction it is to run until the head-end steam valve opens to the proper lead. This should be as follows:

Diameter of cylinder in inches	Lead in inches
10	3/32
12 to 16	7/64
18 to 22	1/8
24 to 26	9/64
28 to 30	5/32

Scribe a line on the shaft and one on

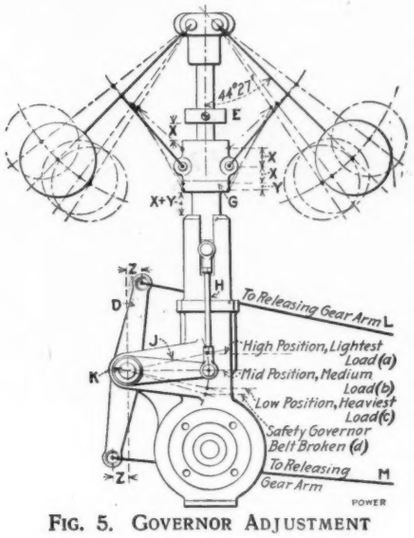


FIG. 5. GOVERNOR ADJUSTMENT

the eccentric hub as a check against slipping and draw up one setscrew in the eccentric sufficiently to prevent its slipping. Turn the crank to the crank-end dead center, level as before and observe the position of the crank-end steam valve. If the hook-rod pin is at the top of the wristplate, the crank-end steam valve should show less lead than was given to the head-end valve, or even a slight lap. If the hook-rod pin is at the top of the wristplate, the crank-end steam valve should show less lead than was given to the head-end valve or even a slight lap. If the hook-rod is connected to the lower part of the wristplate, the crank-end steam valve should show too much lead. Either of these conditions is to be corrected.

Slacken the eccentric setscrew and turn the eccentric in the proper direction to move the crank-end steam valve a distance equal to one-half the required correction. The other half of the correction is to be made by lengthening or shortening the eccentric rod. If the amount of correction required is very small, it may be done by adjusting the length of the connection from the wristplate to the crank-end steam valve. When the valves at each end are adjusted for the proper lead, tighten the setscrews in the eccentric and correct the travel marks on the wristplate bracket. With the crank on each dead center, the exhaust lap should be noted and a correction made to equalize any difference by adjusting the lengths of the connections to the exhaust-valve arms.

There are four important points in the travel of any governor; *a*, the high position, Fig. 5, which trips the valves before they have advanced far enough to open the ports; *b*, the mid position, which is used only for adjusting the positions of the cam arms before locating the cams; *c*, the low position, which draws the cams back to allow the wristplate to rock through its full travel without tripping the valves; *d*, the safety position, which brings the safety cams into operation if the governor belt breaks or runs off its pulleys. The high safety position is for the safe operation of the engine and the low position determines the extreme load-carrying capacity of the engine.

GOVERNOR ADJUSTMENT

To adjust the governor and also the releasing-gear cams on a new engine, block up the governor sleeve *G*, Fig. 5, to a distance $X + Y$ (the dimensions *X*, *Y* and *Z* should be furnished by the engine builder). Adjust the length of the rod *H* so that the lever *J* is horizontal. Set

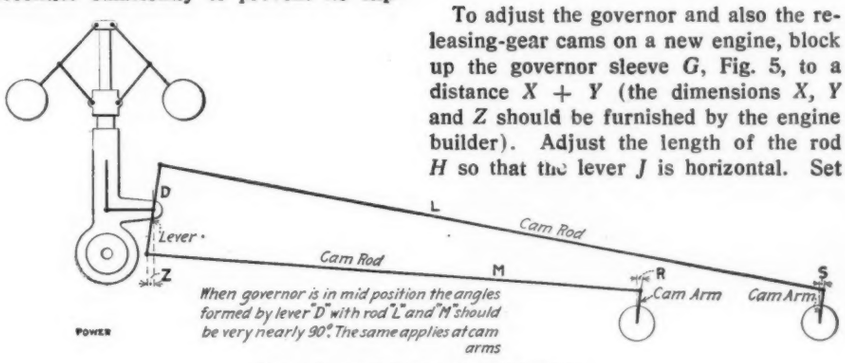


FIG. 6. CAM-ROD ADJUSTMENT

the lever *D* to the dimension *Z* as given on the engine builder's drawing and key the levers to the shaft *K*. Adjust the cam rod *L* and *M*, the proper length to suit the dimensions *R* and *S*, Fig. 6, as shown on the erecting drawing.

Lower the governor from the mid position to the low position. (There should be a device on the governor to hold it up to this position) and turn the wristplate to its extreme travel toward the crank shaft and adjust the cam *J*, Fig. 7, on the head end of the cylinder. This should be slipped around the cam-arm hub *L* until the leveled edge comes to a position 1/32 in. from the hardened-steel block *G* and fasten *J* to *L*. Turn the wristplate to the opposite extreme travel and repeat for the crank-end cam. With the governor in the *low position* the cams should *not* trip the releasing gear.

Next raise the governor to its high position and block it. Move the wristplate slowly and at the same time observe the steam valves. With the governor in this position the cams should trip the releasing gear before the valves come to their opening points. If this condition is not obtained, raise the governor a trifle higher. Allow about 1/16 to 1/8 in. steam lap at the moment of tripping the valve. When this is provided for, bring the collar *E*, Fig. 5, down to the sleeve *G* and fasten.

SAFETY CAMS

To adjust the safety cam *H*, Fig. 7, lower the governor to the low position and block it. Turn the wristplate toward the

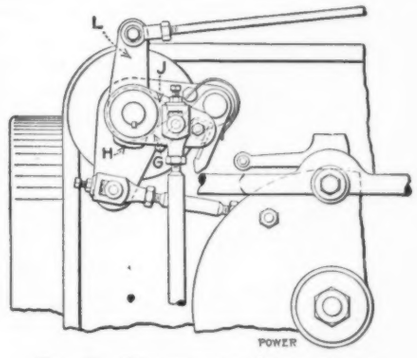


FIG. 7. VALVE-GEAR ADJUSTMENT

head end to its extreme travel. Set the head-end safety cam to within 1/32 in. of the hardened-steel block, Fig. 7. The holes in these safety cams are slotted to allow for some adjustment independent of the cam arm. The fastening bolt should be located to allow for making this adjustment in either direction. To test the safety cams, lower the governor to a position 1/4 in. below the low position and turn the wristplate to its extreme travel in both directions. The releasing-gear hooks should *not* move the steam valves. If the engine is old and not provided with safety cams, they should be added. To allow for this, lower the governor to its very lowest position and then block it up 1/2 in. Let this position be the low position and adjust the cam rods to move the cams to within 1/32 in. of the hardened-steel hook blocks when the wristplate is turned to its extreme limits of travel. Then raise the governor until the steam valves trip at about 1/16 in. lap.

Block the governor in this position and bring the set collar *E*, Fig. 5, down on the sleeve *G* and fasten the set collar to the governor spindle. Lower the governor to the low position 1/2 in. above the very lowest or safety position and adjust the safety cams, as for a new engine. If the valves are carefully set very little, if any, adjustment will be required when the indicator is applied to the engine.

Power Plant of Sectionalized Machinery

By George O. Green

Mule-back transportation is frequently used in mountainous districts to move mining machinery, which is built, if possible, in sections weighing not over 200 lb. Heavier pieces must be carried by gangs of laborers.

Parts of machinery which cannot be made to meet this maximum weight are sometimes packed on a cradle slung between two mules in tandem. Steel cables, weighing 600 lb., are brought in by three mules in tandem. The cables are divided into six coils of 100 lb. each and two coils are placed on each mule, with a short length of cable hanging loose between them.

An extra long and heavy cable was once carried into a Mexican mine by uncoiling it and placing laborers at a distance of 10 ft. apart. At a whistle signal, given by a foreman, each man lifted a portion of the cable and carried it until another signal was given to stop and rest.

The trail leading to the mines of the South American Development Co., near Zaruma, Province of El Oro, Ecuador, is a notoriously bad one. Machinery can be transported by river steamer to Santa Rosa, but for the remaining distance to the mine, 45 miles, everything has to be packed in by either one of the methods mentioned.

The greater part of this stretch of trail is the bed of the Santa Rosa River, which is forded 49 times in a distance of 13½ miles. After leaving the river an ascent of 7000 ft. is made in the course of a three hours' ride.

It is impossible to give any idea of this trail from a photograph, but Fig. 1 shows a bad part. Fig. 2 is a view of a mule arriving at the mines laden with the top section of a steel form used in the construction of a concrete pipe now in course of construction for a hydraulic power plant.

The main part of the plant consists of a 40-stamp mill, three compressors, and a cyanide plant, power for which is obtained from 10 waterwheels under a head of 38.8 ft. There are also pumps, hoisting engines and rock-drills, which are run by compressed air. This machinery was all transported in sections. The water for driving the waterwheels is taken

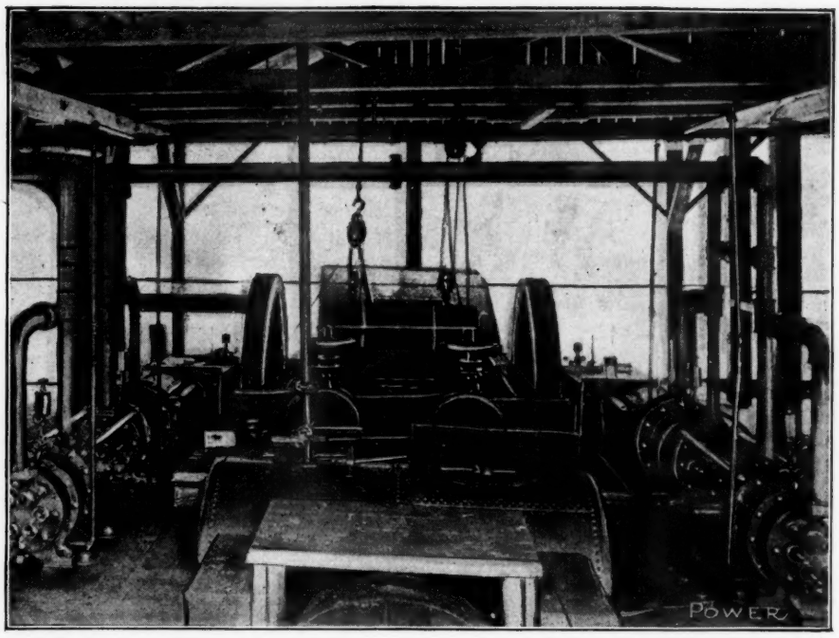


Fig. 5. General View of a Duplex Tandem Air Compressor

Fig. 1. A Bad Piece of Trail

Fig. 2. Mule Arriving at the Mine

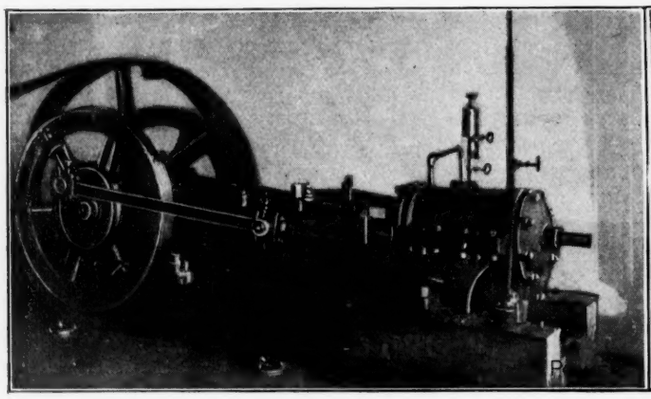


Fig. 3. One of the Air Compressors

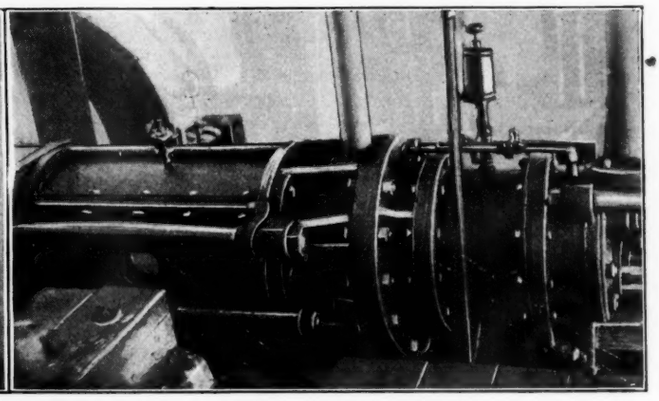


Fig. 4. Sectional Construction

from the Amarillo River through a canal about a mile long.

In Fig. 3 is shown a compressor, an excellent example of sectionalized machinery. Fig. 4 is a view of a cylinder, guide and distance piece of one of these compressors. The distance piece is in two sections; the top and middle piece are plainly shown. The bottom section is similar to the top section. The cylinder is composed of the usual cylinder

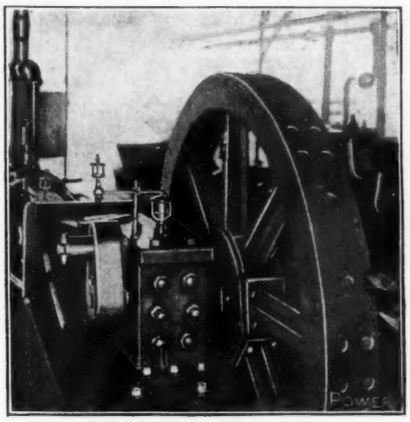


FIG. 6. FLYWHEEL BUILT IN SECTIONS

heads, fitted with poppet inlet and outlet valves, three outer casing sections and a liner. The liner is a driving fit in this outer casing, but is made so as to leave an annular chamber to serve as a water jacket. Fig. 5 shows one of the sectionalized duplex, tandem air compressors.

The flywheel, Fig. 6, is made up of 42 castings. The main bearing and the two top and bottom nuts on the stay rods which hold the main bearing and the guide sections together are also shown. Similar rods on each side hold the front and rear ends of the cylinders together, as in Fig. 4.

One side of the compressor is held in alignment by a steel skeleton frame, embedded in the concrete foundation, with holes drilled to match holes in the cylinder and different sections of the working parts of the machines.

These compressors run at 100 r.p.m., compressing air to a pressure of 100 lb. per sq.in. One has been running for 14, and the other for seven years.

Three natives attend to the lubrication, cleaning, pressure regulating, stopping and starting the compressors. This is about all they are capable of doing, any attempt to teach them to pack piston rods or to take up bearings has proved a failure, but they are good, faithful workmen and easily managed. Their pay amounts to one gold dollar a day.

A clearing house for the issuance of information respecting engineering standards will be established by the American Society of Mechanical Engineers. The prominent engineering societies have been invited to coöperate.

Automatic Motor Controller

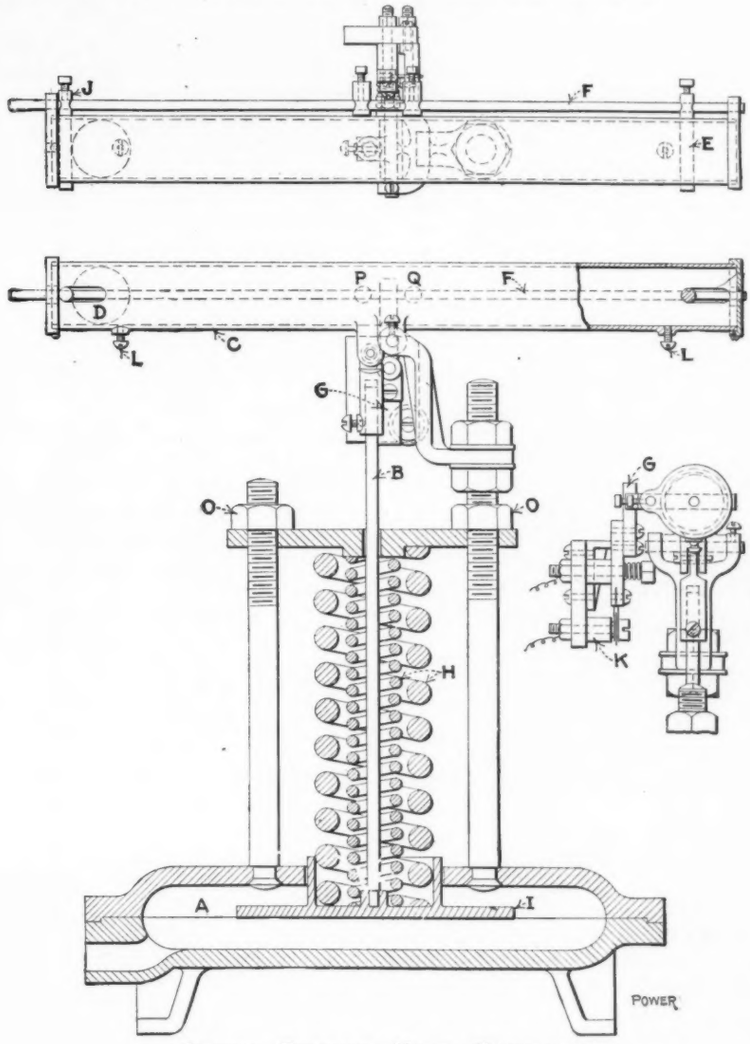
This device is designed to maintain a uniform water level in tanks, standpipes or reservoirs by automatically controlling the action of the motor driving the pump.

The pressure from the tank coming underneath the diaphragm *A* forces the rod *B* upward, tilting the cylinder *C*, when the ball *D* will roll to the opposite end, hitting the pin *E*. The latter being attached to the horizontal rod *F* causes the stop *P* on the rod to move the switch *G*, thus stopping the motor.

When the water level in the tank is lowered, the springs *H* force the diaphragm *A* downward. The rod *B* which is attached to the diaphragm plate *I* tilts the cylinder *C* in the opposite direction, causing the ball *D* to strike the pin *J*. This also being attached to the horizontal rod *F* shifts it in the reverse direction, causing the stop *Q* on the rod to move the switch *G* into contact with the post *K*, which starts the motor.

The adjusting nuts *OO* above the springs *H* regulate the average height of water maintained in the tank, that is, its general level, and the range between the maximum and minimum levels may be varied by turning the setscrews *LL*, shown at each end of the cylinder. These act as obstructions to the free rolling of the ball so that the more they are screwed in the more must the cylinder tilt before the ball can roll over them.

This device is manufactured by the Golden-Anderson Valve Specialty Co., Pittsburgh, Penn.



SECTION THROUGH MOTOR CONTROLLER

The twenty-first annual revised edition of Hendricks Commercial Register of the United States for Buyers and Sellers has just been issued. This edition numbers 1574 pages and contains upward of 385,000 names and addresses.

The value of the Commercial Register for purchasing purposes is not confined to its complete classifiation alone; it also gives much information following the names of thousands of firms that is of great assistance to the buyer, and saves the expense of writing to a number of them concerning the article required.

The book will be expressed to any part of the country on receipt of $10 by the publishers, S. E. Hendricks Co., 74 Lafayette St., New York.

Electrical Department

Conducted to be of service to the men in charge of electrical equipment in the power house

Fan Motors and their Control

By George J. Kirchgasser

In most fan applications it is necessary to be able to vary the speed to meet all conditions of service. Motors for either direct or alternating current can be used where speed variation is required, except the induction motor of the squirrel-cage type. This is satisfactory only where a constant speed is required.

To operate a constant-speed fan of small size it is only necessary to close the switch as shown in Fig. 1. If the motor is above ¼ hp., however, the switch in the fan circuit must first be closed and the motor brought up to speed by means of the starting rheostat. The connections for this purpose are shown in Fig. 2, and a rheostat for use with a direct-current motor is shown in

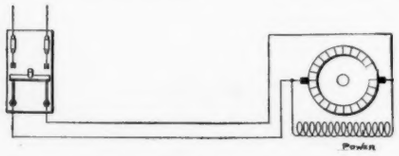


Fig. 1. Connection for Small Constant-speed Motor

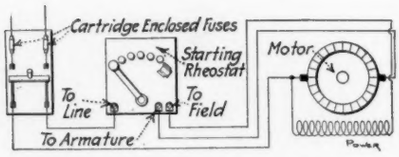


Fig. 2. Connections for Starting Medium-sized Motors

Fig. 3. This type of starter is designed only for accelerating the motor and its resistance is in circuit only during the acceleration period. The speed-regulating rheostat referred to later is similar in appearance but is differently designed.

By referring to Fig. 4 it will be noted that this illustration of a Cutler-Hammer speed regulator is similar to the starting rheostat shown in Fig. 3. The principal difference is in the design of the resistance. For the regulator the resistance must be capable of carrying the full load current for long periods while the resistance of the starting rheostat is only in service at intervals, when starting. To make the action of these regulating rheostats clear a reference to the characteristics of motors may be pertinent.

Where direct current is available a shunt or compound motor may be employed, although the shunt type is used in practically all cases. There are two ways of varying the speed of these

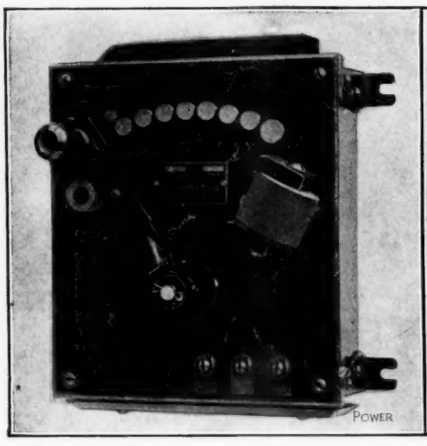


Fig. 3. Standard Direct-current Starting Rheostat

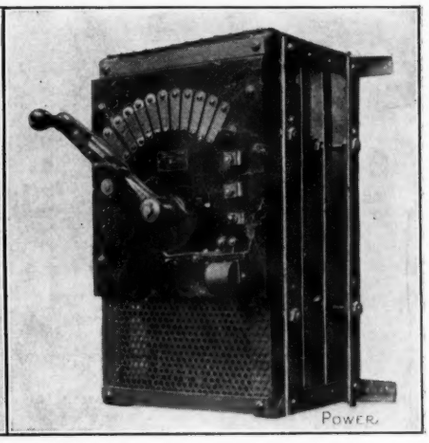


Fig. 4. Hand-operated Speed Regulator

motors; by inserting resistance in circuit with the armature of the motor and by inserting resistance in circuit with the shunt field winding.

Inserting resistance in the armature

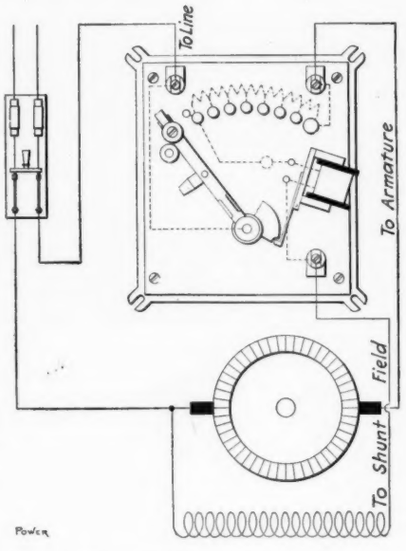


Fig. 5. Regulator Combining Field and Armature Resistance

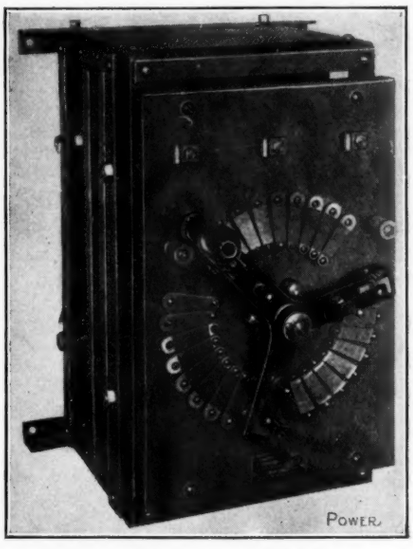


Fig. 6. Regulator for Three-phase Motors

circuit by means of a regulator, such as shown in Fig. 4, reduces the speed of the motor in steps from its normal running speed to somewhere near one-half speed, usually. The amount of speed reduction depends upon the amount of resistance, the position at which the movable lever of the regulator is located and also to a certain extent upon the load the motor is driving.

When resistance is connected in various amounts in the shunt field circuit the current flowing in this field is decreased and causes the motor to increase in speed above normal, the percentage increase depending upon the amount of resistance used. Sometimes regulators are used which combine the two systems, that is, they permit of reducing the motor speed below normal by

means of armature resistance and provide for increasing the speed above normal by insertion of resistance in circuit with the shunt field of the motor. A regulator of this type is represented in Fig. 5.

Regulators used with motor-driven ventilating fans or for similar service are referred to as doing "fan duty." This is to distinguish from "machine-duty" regulators in which the resistance is differently proportioned. A "fan-duty" reg-

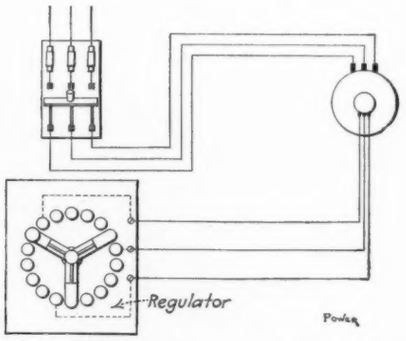


FIG. 7. CONNECTIONS FOR THREE-PHASE MOTOR REGULATORS

ulator is used in service where the torque decreases as the speed decreases, as is the case of a motor-driven ventilating fan. The current required when running at a speed less than the normal is less than the current required when running at the normal speed. For this reason the ventilating fan offers one of the best applications for control by means of an armature resistance type of regulator

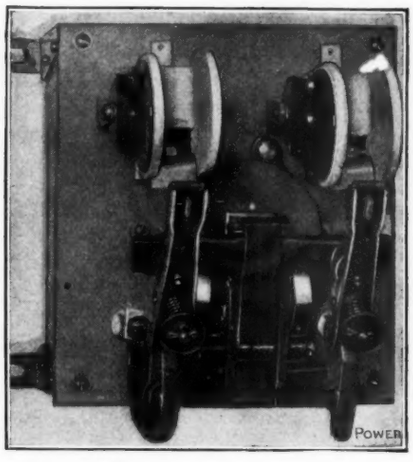


FIG. 8. MAGNETIC SWITCH STARTER

because at low speeds it is not necessary to dissipate a large part of the current in heating the resistance.

With many classes of service this is not the case, and full current is required at half speed, three-quarter speed or full speed, the motor running at the lower speeds simply because a part of the energy is dissipated in the form of heat.

Single-phase and polyphase motors are often used for driving fans, the former usually for the small sizes. Of the polyphase motors those of the squirrel-cage type are used only in cases where a constant speed is satisfactory. Where variations in speed from 50 per cent. below normal to normal are required and alternating-current supply is provided, the slip-ring type of motor with a special regulator is well suited. Fig. 6 shows such a regulator commonly known as the faceplate type, which has three sets of contacts. The steps of resistance for varying the speed can be inserted in circuit with the phase windings of the rotor, by means of a single lever operating the three sliding shoes. Equal amounts of resistance are cut in or out of circuit in the three phases (or two phases if a two-phase motor), thus maintaining a balanced running condition. The connections in Fig. 7 show how this resistance is cut in and out of circuit.

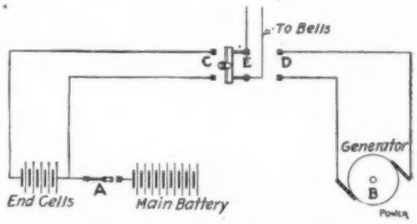
The drum-type regulators, similar in appearance to street-car controllers, are well adapted for larger fans, and are particularly suited for fan systems used in grain and malt elevators, in textile mills and wherever flying dust or other particles are present. Being entirely inclosed, nothing can come in contact with the current-carrying contacts.

In some fan installations it is desirable to control the starting and stopping, and sometimes the speed, from a point distant from the fan, such as another floor of the building. For the starting and stopping a magnetic switch starter, such as shown in Fig. 8, which can be actuated from push-button stations, is suitable. But in hotels, municipal buildings, schools, etc., where the various rooms are used at different times for different purposes, an automatic fan controller arranged so that from each room or group of rooms the speed of the fan can be changed, is of particular advantage. Control equipments giving these features are being installed to a large extent.

CORRESPONDENCE

End Cells for Bell Circuits

In plants having a storage-battery system, much annoyance and expense can be avoided by utilizing the end cells for operating elevator signals, call bells, annunciators, house phones, and any other apparatus requiring a low voltage. The number of cells that are required to make up the necessary voltage should be detached from the main battery as in the sketch. They should then be connected with the main battery by a single-pole switch of sufficient capacity to safely carry the full charging current. This detached group of cells is then connected to one side of a double-pole, double-throw switch as at *C*. The other side of the double-pole switch *D* is connected to a small motor-generator *B* and the middle point *E* to the main feeder of the low-voltage system. The object of the heavy single-pole switch between the end cells and the main battery is to prevent a high voltage backing up into the bell system should a ground occur.



BELL CIRCUIT ON BATTERY

When it is necessary to charge the end cells, the motor-generator is started and is connected with the bells by means of the double-throw switch. This consequently disconnects the end cells from the bell system.

The heavy single-pole switch at *A* can then be closed and charging can be continued as long as is necessary. The only object in having a motor-generator is to prevent interruption in the bell system, during the charging of the end cells.

FREDERICK HANSA.

New York City.

Blown Fuse

Mr. Piper's answer to Mr. Koppel's question which appeared in POWER for Nov. 5 is undoubtedly true, but the conditions stated by Mr. Koppel cannot be fulfilled in any three-phase system. Evidently the three voltage determinations were not made simultaneously or under identical conditions. This fact is easily proved by taking three space vectors, one of length equal to two, one of length equal to one, and one of zero length; upon trying to arrange these vectors in any manner it is readily seen that between three points the condition of relative potentials of two, one and zero is utterly impossible.

HENRY E. RANDALL, JR.

Boston, Mass.

Cleaning Battery Jars

Many engineers using storage batteries with glass jars have probably had considerable trouble keeping them free from spray and dirt.

When my jars get dirty, I wipe them off with a piece of waste moistened in kerosene. The oil softens the dirt and leaves the jars with a greasy surface, which will not allow the spray to stick. Care should be taken to get off all superfluous oil, as that would gather dust.

H. K. WILSON.

New Bedford, Mass.

Gas Power Department

Worth-while gas-engine and producer information treated in a way that can be of practical use

A Bituminous Gas Producer Plant

By A. A. Potter

The producer plant here described was installed in 1908 by the Smith Power Co., of Lexington, Ohio, as part of the laboratory equipment of the Kansas State Agricultural College. As this institution is far from the natural-gas fields of Kansas and was not near an artificial gas plant, the producer gas had to be adopted for use in open burners at the various laboratories.

Plant Equipment

The producer plant is shown in elevation and plan in Figs. 1 and 2 respectively. The essential parts of this plant are a gas generator *B*, condenser *H*, centrifugal wet scrubber *E*, centrifugal fan *K*, gas-pressure controlled reducing valve *Q*, tar-collecting tank *W*, dry filter or purifier *V*, gas holder (not shown) and pipe lines.

Details of the generator are shown in Fig. 3. It consists of a steel shell 6 ft. in diameter and 10 ft. high, lined with one course of firebrick and one course of ordinary brick, making a total thickness of about 9 in. The diameter of the fuel bed is 54 in., and, assuming a gasification of 10 lb. of coal per square foot of fuel area this producer is capable of gasifying 159 lb. of coal per hour and will generate about 10,000 cu.ft. of gas. The heat of combustion of the gas made is about 150 B.t.u. per cu.ft. (high value). Fuel is charged into the retort *C* and is admitted to the shell of the generator by means of a quick-opening gate valve. The retort *C* is provided with a water-sealed cover, this arrangement enabling the operator to charge the producer while the plant is in operation, without the danger of admitting air or of allowing gas to escape. Coal entering the shell is distributed by means of the hood *J'*, the inside of which serves as a gas collector; the gas outlet is at *J*. A swinging grate *Z* supports the fuel bed and is suspended from the shell by four chains, the shaking motion of the grate being produced by the hand lever *L*. Doors are provided at the bottom for the removal of ashes. The generator is also arranged with poke holes on top and with peep holes at the various zones, so that the temperature and condition of each zone may be determined. The inlet for the steam and air is shown at *A*.

Gas leaving the generator at the outlet *J* enters the condenser, which is shown in cross-section in Fig. 4. The condenser consists of a number of deflecting rings which throw the gas and water alternately from side to center, cooling the gas and condensing part of the volatiles.

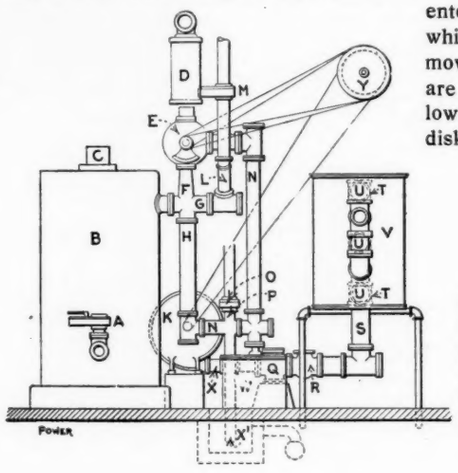


Fig. 1.

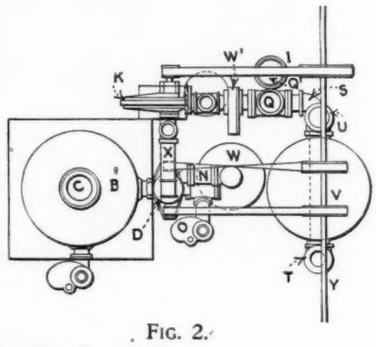


Fig. 2.

Elevation and Plan of Producer Installation

Referring again to Fig. 1, the gas, water and impurities, after leaving the condenser *H* are separated in the tee at *N'*; the gas is carried upward through the 6-in. pipe to the scrubber *E*, while the water and the condensed volatiles fall to the tar-collecting tank *W*. The water from this tank is allowed to pass off into the sewer, while the tar settles to the bottom and is drawn off.

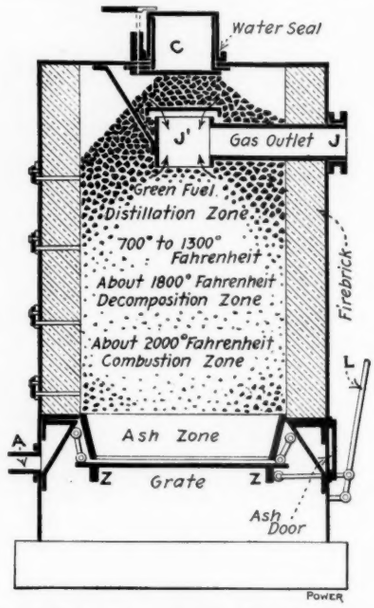


Fig. 3. Section through Gas Generator

The gas passing up the 6-in. pipe *N* enters the centrifugal wet scrubber *E* which is shown in detail in Fig. 5. The moving parts of the centrifugal scrubber are two disks *T* and *T'*, carried by hollow shafts. About the peripheries of these disks are blades or vanes ½ in. wide and spaced ¼ in. apart. The disks are of different diameters, so that one is inside the other, and are driven in opposite directions by open and crossed belts. The gas enters the center of the scrubber at *S'*, is thrown by centrifugal force to the circumference of the disks and in passing between the blades is intimately mixed with water which is admitted through the hollow shaft *R*.

Leaving the centrifugal scrubber the gas passes up to the separating drum *D* (see Fig. 1), where its velocity is decreased and part of its moisture is removed. It is then led through the pipe *X* to the centrifugal fan *K*. The fan *K* produces a suction on the fuel bed and also pushes the gas through the pipe *X*, the gas-pressure controlled reducing valve *Q*, the check valve *R*, and the pipe *S*. The function of the reducing valve *Q* is to regulate the pressure and amount of steam passing into the gas generator at *A*.

From the pipe *S* the gas is admitted to the purifier *V* through the three inlets *U*. This type of purifier, sometimes called a dry filter, is used for the separation of sulphur from the gas, and is shown in cross-section in Fig. 6. It consists of a steel shell having three inlets and two outlets. Inside of the tank are four wooden lattice-work shelves, supported on rings attached to the shell. Four-inch

layers of iron sponge are placed on each of these shelves. This iron sponge (iron oxide) is prepared by mixing iron filings,

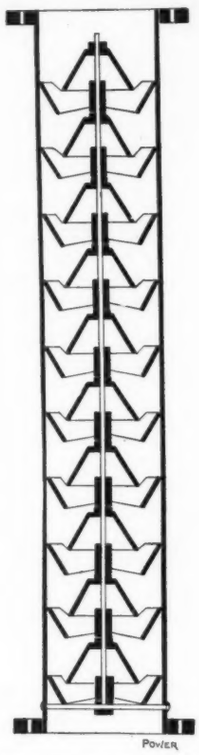


FIG. 4. SECTION THROUGH CONDENSER

sawdust and sal ammoniac. The fan *K* forces the gas through the filter frames in the purifier; the gas leaving the purifier through the outlets *T* passes through the venturi-tube meter to the gas holders and mains.

The fan and centrifugal scrubber are plant is provided with a Junkers constant-pressure gas calorimeter for the determination of the heat of combustion of the gas made, scales for weighing the coal and ashes, a water meter for the measurement of the scrubber water, a steam meter for the determination of the steam used in the process, manometer tubes and thermometers at the various points in the plant, a Heraeus pyrometer for the determination of the temperatures in the various zones of the producer, and electric meters for the measurement of the power required to drive the centrifugal fan and wet scrubber.

CARE AND OPERATION OF PLANT

The general operating details can be seen by referring to the weekly log sheet. (Fig. 7.)

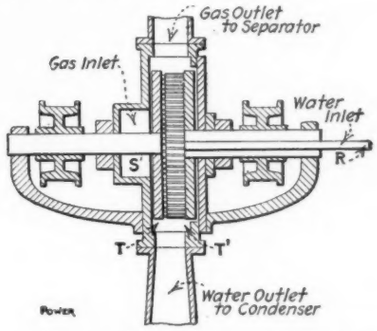


FIG. 5. CENTRIFUGAL SCRUBBER

In the operation of a gas-producer plant there are two processes, the blowing and the gas-generation process. Starting with the producer idle, the valve *M*, connecting the fuel bed with the atmosphere, will be found open. The first step is then to work the fuel bed with a

each shipment of coal is taken and this is quartered down to an amount sufficient to fill a two-quart can and sent to the laboratory for analysis. After the gen-

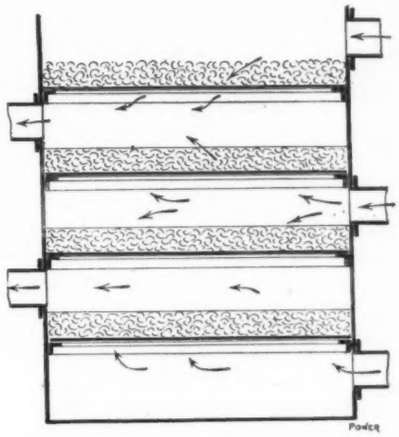


FIG. 6. SECTION THROUGH PURIFIER

erator is filled with coal, the water to the scrubbers is turned on, the fan is started, the air valve *A* and the purge valve *O* are opened, and the valve *M* is closed. This starts the blowing process, which requires 10 to 30 min., the ashes are then shaken down, the clinkers are pulled from the grate and the ashpit cleaned. The electric meters are then read and the flow of water to the scrubber is adjusted to the most economical rate, which with this size of producer has been found to be about 1 cu.ft. per min. When the fuel bed appears hot as viewed through the peep holes of the generator, steam is turned on, and the purge valve is throttled down sufficiently to produce a slight pressure on the pilot

WEEKLY LOG OF GAS PRODUCER _ RATED CAPACITY 100 HORSEPOWER

Run No.	Date	Gas Generation: Time From	To	Total	Current: Volts	Amps	Kw.H.	Scrubber Water: Meter Start	Stop	Cubic Feet	Steam: Gage Pressure	Amount Pounds	Gas: Average Flow	Total Cubic Feet Generated (Corrected	Pressure	Calorific Value: High	Low	Coal: Lot Number	Weight Pounds	Ashes Pounds
1																				
2																				
3																				
15																				
16																				
	Total																			
	Average																			

Run No.	Temperatures: Degrees Fahrenheit: Scrubber Water: Initial	Final	Rise	Gas: Producer Outlet	Scrubber Outlet	Per Cent. Combustion Zone Producer	Heating Fuel Bed During Blowing Process: Time: From	To	Total	Current: Volts	Amperes	Kw. H.	Scrubber Water: Meter: Start	Stop	Cubic Feet	Remarks	Operator
1																	
2																	
15																	
16																	
	Total																
	Average																

POWER

FIG. 7. WEEKLY LOG SHEET

driven by a 13½-hp. electric motor. There is no boiler or vaporizer with this plant, the steam for the producer being supplied from the main boiler plant of the college.

Besides the above equipment, this poking bar, so as to settle the coal in the generator. Coal is then hoisted to a platform which is lowered by means of a hand lever on a weighing scale, and the total weight of coal required to fill the producer is recorded. A sample from lamps located at the fan outlet. If the gas flame at the pilot lamp indicates that the producer is generating a gas of the proper quality, the purge valve *O* is closed and the valve *Q* is opened, allowing the gas to pass through the dry

purifier and the venturi meter to the gas holders.

In stopping the plant the motor switch is opened, the steam is shut off, the valve at the purge pipe *O* opened, the valve *Q* leading to the purifier closed, the flow of water to the scrubbers is stopped and the natural-draft valve *M* opened. The air valve at *A* and purge-pipe valve *O* are then closed and the producer is left in this condition ready for starting.

To the foregoing directions for starting and stopping may be added—if the producer remains idle for any length of time it may be necessary to start a fire in the fuel bed. If the bed contains considerable fuel this can be best accomplished by using gas to ignite the coal. If the producer is to be started for the first time or the bed is thin, a fire can be started in the usual manner and the fuel bed built up.

The operator, besides following out the directions as outlined, records on the weekly log sheet (Fig. 7) the time required for the blowing process, the power necessary to drive the auxiliaries during that time in kilowatt-hours, and the water used in the scrubbers. When the blowing process is completed and the gas generated is admitted to the holder and pipe lines, the operator takes the various readings shown on the weekly log under "gas generation." This includes the time the producer is generating gas, the power required for driving the auxiliaries, in kilowatt-hours, the amount of cooling water used in the scrubbers, the steam necessary for the process, the rate of gas making, the cubic feet of gas made, and the gas pressure. Also one analysis of the gas is made and the temperatures of the gas in the combustion zone as well as at the producer and scrubber outlets are taken.

It has been found that the operator can attend to the operating details of the plant and have sufficient time to take the readings and keep up the records.

From the weekly records calculations are made of the cubic feet of gas generated per pound of coal, the cost of fuel, water, steam, electricity and labor per thousand cubic feet of gas and per thousand B.t.u.

These records and the results of the coal analyses constitute a mass of valuable information regarding the operation of this type of prod cer. It is the intention to publish these results in bulletins of the engineering-experiment station of the Kansas State Agricultural College. It has also been found that the taking, keeping and working up of complete records are conducive to better work on the part of the operator.

On this page Mr. McGouvy speaks of gas-engine builders who are willing to guarantee a horsepower-hour on 8300 B.t.u. We have yet to meet such reckless builders.

Gas Power Don'ts

By F. Webster Brady

Don't expect a horsepower from a half-pint of gasoline.

Don't buy an engine too small for your work.

Don't use galvanized-iron pipe for the oil connections.

Don't put a rubber gasket under the cylinder head.

Don't use baling wire for the battery connections.

Don't foul the vent hole in the gasoline tank.

Don't use butterine in the bearings.

Don't test the batteries by holding a screwdriver across the terminals.

Don't climb into the flywheel to start the engine.

Don't tramp on the oil pipes and igniter wires.

Don't forget to drain the water jacket the night before Xmas.

Don't kick up a dust about the carburetor.

Don't fail to strain all the gasoline.

Don't set the timer at "advance" before cranking.

Don't use rubber hose for gasoline connections.

Don't pour water into the carburetor.

Don't neglect the strainer in the oil pipe.

Don't expect air cooling to be automatic.

Don't crank with a push and with the right hand.

Don't cuss the engine 'cause it smokes.

Don't buy an engine without adjustable bearings.

Don't leave the engine under the rain spout when laying it away.

Don't trust an inexperienced man to operate your engine.

Don't dismantle the magneto simply to gratify an idle curiosity.

Don't operate without a muffler just to spite your neighbors.

Don't forget that not all engine pounds are caused by loose parts.

Cleaning Blast Furnace and Coke Oven Gases

In the *Proceedings* of the Société de l'Industrie Minérale, M. A. Gouvy emphasizes the importance of cleaning blast-furnace and coke-oven gases before they are used even for burning under boilers. Cleaning the gas so that it does not contain more than 0.5 gram per cu.m. (0.218 grains per cu.ft.) greatly increases the evaporative power of any boiler heated with it. With the unclean gas about 2 cu.m. are required per kilogram (32 cu.ft. per lb.) of water evaporated as compared with only 1 to 1.2 cu.m. per kg. (16 to 19 cu.ft. per lb.) with clean gas. The effect of cleaning on the evaporative efficiency is due to the fact that the dust is an excellent nonconductor and if present is deposited upon the heating surfaces, greatly impeding the transmission of heat through them to the water. In a Westphalian works where the cleaning plant reduces the dust contents to less than 0.25 gram per cu.m. (0.109 grain per cu.ft.) the consumption of gas is only 0.8 cu.m. per kg. (12.8 cu.ft. per lb.) of steam superheated to 250 deg. C. (482 deg. F.).

The dust deposit has another prejudicial effect in acting as a flux and making the firebrick waste rapidly. The cost of the cleaning should not exceed 0.027 franc per 100 cu.m. (0.15c. per 1000 cu.ft.), including all charges.

For engine running the gas should be further purified so as to bring its dust content below, as a maximum, 0.03 gram per cu.m. (0.0132 grain per cu.ft.) and in favorable conditions this figure has been reduced to 0.005 gram per cu.m. (0.002 grain per cu.ft.), in which case the engines do not require to be stopped for cleaning purposes any more than do steam engines. Gas-engine builders will generally guarantee a consumption not exceeding 2100 cal. (8300 B.t.u.) per i.hp.-hr., and in practice it may be taken that the consumption will not exceed 2800 cal. (11,100 B.t.u.) per b.hp.-hr., which with blast-furnace gas having a calorific value of 950 cal. per cu.m. (107 B.t.u. per cu.ft.) corresponds to a consumption of about 3 cu.m. (106 cu.ft.) of gas per b.hp.-hr. This is really a maximum value, but a low gas consumption is of secondary importance as compared with reliability.

Preheating for Internal Combustion Engines

On page 565 of the Oct. 15 issue an article appeared on "Preheating for Internal Combustion Engines." I do not want to dispute this matter for I have been working on this subject for several years. I have had patents pending in the United States Patent Office covering this field for some time. What my patents cover is: "The field of obtaining ignition of the fuel by means of the heat of compression added to an initial preheat." This preheat can be obtained by the heat of the exhaust or by an external heater, depending upon the type of the engine required.

I am not a wealthy man and, considering the expenses of experiments, etc., have not made as much progress as I should like to have made. However, I have worked quite a way beyond the scope of the article in question and I ask you to publish this so that if any of your readers are taken with this idea of preheat they will not infringe on my patents unwittingly.

John F. Wentworth.

Quincy, Mass.

Refrigeration Department

Principles and operation of ice-making and refrigerating plant and machinery

Economic Operation of Ice Making Machinery by Use of Thermometers*

By V. R. H. Greene

In practical refrigeration the gases most commonly met with are CO_2 and ammonia. Ammonia is used more extensively because it can produce the refrigerating effect at a lower pressure for the same water temperature and because in the evaporation of one pound of the liquid to a gas a larger number of heat units can be absorbed. The cycle of the ammonia gas in an ordinary ice plant is as follows:

Ammonia gas is drawn from the ice-tank coils through the suction valve into the gas pump or compressor. On the return stroke of the piston the gas is compressed and discharged under pressure into the ammonia condenser, where the heat of liquefaction is removed and the ammonia gas reduced to liquid. This liquid is drained from the condenser pipes to a receiver, which is under the same pressure as the condenser, and on this account the ammonia gas remains in a liquid state. From the receiver the liquid flows to the freezing coils, but the supply is regulated by a delicately adjustable valve. Once in the tank coils under reduced pressure the liquid changes to a gas and in so doing absorbs from the brine nearly as much heat as was taken up by the water flowing over the ammonia condenser.

In any vapor, temperature of liquefaction and the corresponding pressure bear a fixed relation to each other, and the temperature of the ammonia in the freezing-tank coils depends upon the extent of reduction of pressure in the low-pressure side of the system. Just as it is necessary to have two different levels to make water flow from one vessel to another, so there must be two different temperatures to cause a transfer of heat from one body to another. In a freezing tank the most economical brine temperature that can be maintained is 12 to 14 deg. F., and to produce such a temperature the temperature of the ammonia gas in the freezing coils must be about 0 deg. F. This 12- to 14-deg. difference between gas and brine is required to cause the necessary heat transfer from the brine to the liquid ammonia in the coils.

*Abstract of paper read before the Eastern Ice Association, New York City.

Any table giving the properties of ammonia gas will show that an ammonia temperature of 0 deg. F. corresponds to a pressure of approximately 15-lb. gage. This is why a back pressure of 15 lb. is most commonly met with in ice-making practice.

Refrigerating effect is produced wholly by the evaporation of a given number of pounds of liquid ammonia into a vapor. This vapor is then drawn into the compressor and compressed and discharged into the condenser. As there are more pounds of ammonia contained in a cubic foot of its vapor under a high pressure than under a low pressure, it is advisable for the economical operation of the plant to carry the suction pressure at the tank as high as possible consistent with the brine temperature required. With a high back pressure the more closely are the gas molecules compressed, and, therefore, the more pounds of ammonia are introduced into the compressor cylinder and discharged into the condenser per cubic foot of vapor handled by the compressor.

However, a high suction pressure means a high suction-gas temperature, and this tends to reduce the degrees difference between the brine and gas, resulting in a higher brine temperature. As it is uneconomical to make ice with a brine temperature much higher than 14 deg. F., owing to the length of time required to freeze a standard ice can, some method must be devised to keep the brine bath at 14 deg., and at the same time make it possible to run with as high a back pressure or suction temperature as possible. It is known that a greater rate of heat transfer is obtained between two liquids or gases in motion than at rest, and to help increase this rate freezing tanks are provided with agitators.

The first step, then, toward freezing-tank efficiency is to operate the tank agitator as fast as possible without causing the brine to slop into the ice cans. The next step is to place thermometers in the suction pipe near the entrance into the compressor cylinder and find the smallest degree difference which it is possible to carry and maintain a brine temperature of 12 to 14 deg. F. When this temperature is found, regulate the operation of the plant thereafter by the thermometer, paying attention to the expansion valve regulation only to maintain this suction temperature at the machine. If the temperature is right the suction pressure must be right, for unless the gas is superheated it will correspond to that pressure which is equivalent to the suction temperature maintained.

There is another loss which the thermometer on the suction eliminates. As previously stated, under a pressure of 15 lb. saturated ammonia vapor has a temperature of 0 deg. F. It is possible, however, to maintain a pressure of 15 lb. at the freezing tank and then allow the suction gas, after it leaves the tank on its way to the compressor, to pass through hot engine rooms in uninsulated pipes and become heated up away beyond the temperature corresponding to its pressure. When this happens the gas expands under heat, just as do other substances, and becomes rarefied, so that for every cubic foot of gas pumped by the compressor very few pounds of ammonia are delivered to the condenser per cubic foot of gas pumped. This results in an overspeeding of the ammonia compressor in proportion to the superheat in order to discharge the required number of pounds of ammonia into the condenser. By regulating the temperature of the gas at the point of entrance to the compressor, this loss is eliminated.

The only possible loss of efficiency that can result from thermometer operation in the suction side of the machine, is caused by carrying too much liquid ammonia back from the freezing tank into the compressor cylinder, making a semi-liquid pump instead of a pure-gas pump. When this is done the compressor displacement is spent upon removing the liquid which has become evaporated into a vapor in the compressor itself, and not the vapor which has done useful refrigerating work in keeping down the brine temperature. This will show at once a capacity loss, which can be discovered by the drop in temperature of the discharge gases to the condenser. No liquid ammonia should be allowed to enter a compressor cylinder.

When vapor containing no liquid ammonia is compressed from a suction pressure of 15 lb. to a condenser pressure of 185 lb., which is common in practice, a temperature of about 250 deg. F. is created. This temperature is based only upon the vapor entering the compressor at the temperature corresponding to the back pressure of 15 lb., which is 0 deg. F. If the thermometer is placed in the discharge pipe near the compressor and reads about 250 deg. F., it will indicate that the entering vapor temperature

is the right one, and that no liquid is present in the compressor. Should the discharge temperature go up to 270 deg. F., it will be known at once that the suction gas is entering the compressor at a temperature above zero, and is therefore superheated and rarefied. This shows that the expansion valves on the freezing tank are becoming clogged and not enough liquid is being fed into the tank coils. If, on the other hand, the discharge temperature falls below 250 deg. F., too much liquid is passing into the coils, meaning that the compressor is becoming a liquid and not a gas pump, and that this flow must be checked. It will then be necessary to pinch down upon the expansion valve which appears to be feeding in excess of all the others, until the proper discharge temperature is reached.

In a 30-ton plant which seldom made over 20 tons a day, the compressor was overhauled, new valves installed and excessive expense entailed with no increase in output. When a thermometer was placed in the discharge line at the compressor a temperature of only 104 deg. F. was registered, proving excessive ammonia fed into the tank coils. The expansion valves were closed down until the discharge temperature rose to 240 deg. F., which proved to be the best temperature, and from that time on 32 tons of ice per day have been pulled from this plant. Nothing more was done to the machine than to raise the discharge to its proper point for efficient operation.

From the gas compressor the ammonia is discharged to the condenser, which usually consists of a series of 2-in. pipe coils into which gas enters at the bottom and passes upward in counter-current flow to the cooling water, which flows downward over the outside of the pipes. Just as there was required a difference in temperature between the ammonia gas and the brine in the freezing-tank coils to produce the necessary flow of heat, so in the ammonia condenser the water which is showered over the outside must be cooler than the condensing vapor on the inside. In average practice this difference is between 10 and 15 deg. F., so that with an initial water temperature of 70 deg. F. the liquid might be expected to have a temperature of 80 to 86 deg. F., depending upon the type and efficiency of the ammonia condenser.

The same ammonia table which showed that for a back pressure of 15 lb. the saturated gas temperature was 0 deg. F., also shows that for an ammonia temperature of 80 deg. F. a condenser pressure of 138 lb. must be expected, or for a temperature of 85 deg. F., 152 lb. If the actual head pressure as registered by the gage is higher than that corresponding with the liquid temperature, provided, of course, that the gage reading is correct, it will be known that the system is foul with noncondensable gases which should be removed at once so as to reduce the head pressure to the proper point and thus reduce the horsepower required to compress the gas. In an efficient design of ammonia condenser, particularly one in which a strictly counter-current principle is involved, the water heats up as it passes over the ammonia coils, and the temperature of the outlet water corresponds closely with the liquid temperature. When this is the case, it is only necessary to place a thermometer in the overflow pipe in the condenser pan to learn the temperature of the liquid ammonia.

Of late manufacturers have placed on the suction and condenser pressure gages, directly over the pressure dial., a series of red figures giving the corresponding temperatures. This does away with the necessity of referring continually to the ammonia tables. It is only necessary to see that the outlet water from the condenser pan checks up with the red figures over the pressure-gage line for the correct head pressure, and that the temperature corresponding to the suction pressure on the gage checks with the suction thermometer at the cylinder of the compressor.

Refrigerating Problems

One of our readers in Suva, Fiji Islands, recently asked the following questions, which with their answers may be of general interest:

1. In cold-storage rooms, what are the practical objections, if any, to the perforated floor and ceiling system of circulation? Where the air is cooled by a battery of coils, what is the best arrangement to avoid contamination of the goods stored?

2. For a daily output of three tons of ice, what would be the approximate size of a brine tank, the cans being 8x15x28 in. and the bath temperature 25 deg. F., with a forecooler in use?

3. What would be the approximate size of a coil room 7 ft. high and of an ammonia compressor to maintain 10,500 cu.ft. of space at an average temperature of 25 deg. with the atmosphere at 85 deg. F. and circulating water 70 deg. F., assuming the insulation to be fairly good?

1. Perforated floors and ceilings for circulating the air in cold-storage rooms are very rarely used in modern construction for these reasons: Perforations in the floor weaken it and good circulation of air is only secured when the perforations are so placed that the goods cannot cover them. It is also difficult to proportion the number and sizes of perforations to obtain an even distribution of air throughout the room. Perforations in the floor are also undesirable on account of the dirt which falls through them and collects in the air ducts. Whenever refrigeration is secured in cold-storage rooms by mechanically circulating the air from a central battery of cooling coils, the air is purified at the point where it is cooled, as it deposits its moisture there and the condensed water vapor contains most of the foul gases in solution. Where the "Linde" system is used for cooling, the brine flowing over the outside of the cooling coils purifies the air, but in this case sometimes the calcium is carried along with the air and deposits on the goods that are stored in open crates or are loose. If the air is properly divided during cooling, either the dry-coil or wet-coil method of cooling should purify the air sufficiently for cold-storage purposes. If it does not, any amount of filtering will not improve its quality.

2. A brine tank to hold a sufficient number of 100-lb. cans, 8x15x28 in., to make three tons of ice per day when operated 24 hr., the temperature of brine being 25 deg. F., will be about 10 ft. wide, 24 ft. long and 30 in. high, inside measurements. There will be required about 150 ice cans of the size above mentioned and in the tank should be placed not less than 300 ft. of 1¼-in. pipe per ton of ice-making capacity. To obtain this output with the above tank, the brine must be properly agitated by means of a propeller placed at one end of the tank and the tank partitioned lengthwise through the center to facilitate circulation.

3. A cold-storage room of 10,500 cu.ft. volume lined with an insulation permitting not more than 3 B.t.u. heat transfer per square foot per degree Fahrenheit difference of temperature per 24 hr. will require 2½ tons actual refrigeration to maintain it at 25 deg. with a temperature of 85 deg. F. on the outside. In this calculation no refrigeration is included for materials to be cooled or frozen in the room as no quantities are given. For determining the number of feet of 2-in. pipe necessary in a coil battery to refrigerate this room by cold air, mechanically circulated, it can be assumed that 100 ft. of 2-in. pipe will take care of one ton of refrigeration provided the air passes the pipes of the cooling coil at a speed of about 1000 ft. per min. For 2½ tons of refrigeration, 250 ft. of 2-in. pipes will be necessary under the above conditions. Two cooling coils, each containing the required length of 2-in. pipe should be installed when the plant is continually operated so as to allow for defrosting one coil when the other is in use. To the above must be added the refrigeration necessary to cool and freeze whatever goods are to be stored in the room at 25 deg. F., if they are not already cooled and frozen.

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POWER

Pipe Line Inspection

A factor, closely allied with power-plant safety, that has been ignored is the inspection of steam pipe lines.

Boiler inspection is maintained in several states and in many cities, both in compliance with an inspection law, and also by insurance companies as a business. Why then stop with the boiler? Why not know that the steam lines are correctly designed, installed and cared for?

It is true that fewer accidents occur from steam pipe than from boiler failures, but there are more of them than would occur if the piping were put up right and kept in condition.

When a power plant is installed, in some instances the boilers are inspected before leaving the shop, by a representative of the purchaser, to guard against inferior material and workmanship. They are again inspected before a boiler-insurance company will issue a policy covering the risk against explosion. This is as far as the inspection goes. The steam piping can be put up in any manner that suits the designer or contractor. It may contain pockets, dangerous bends and cast-iron fittings unprotected by suitable means of caring for expansion and contraction strains.

When such piping is once in place it is there to stay until an accident occurs. It may be the cracking of a fitting, the fracture of a blank flange that is altogether too light, or it may be the pulling apart of a screwed joint. Anyone of these could be avoided if the pipe work were under the eye of an inspector. Starting right and keeping right means a lot in any work, and especially in power-plant construction.

A second demand for inspection occurs where the piping was originally designed for a pressure of, say, one hundred pounds, and which is later forced to carry a pressure of a hundred and fifty pounds. This is the same as using standard weight pipe instead of extra-heavy pipe; it is not an uncommon practice.

There are engineers operating plants, the high-pressure piping of which they know to be unsafe, because it was never designed to carry the pressure that has been forced upon it.

It is seldom that a piece of pipe line is removed because it is considered unsafe. Joints may be found leaking badly, the drainage may be faulty or none at all provided, and the means of suspension may be such that the pipe and fittings are always under a strain, yet these conditions will generally be allowed to continue.

Because steam pipe lines are small in diameter and are considered safe for any pressure to which they will ever be subjected, is no reason for confining the inspection to the boilers, and losing sight of the dangers existing beyond the stop valve. There is no excuse for neglecting the inspection of steam lines at intervals. If steam-piping inspection were made a part of the power-plant routine fewer accidents would occur, and if this had been the practice aforetime many men now dead would be living.

Reporting Plant Conditions

Under the stress of operating service a change in shifts in the power house sometimes represents a critical period in the daily routine. As one set of men lay down their tasks and another set assumes them, it is clearly desirable to make the transition as smooth as possible and to leave no loose ends or forgotten duties to cause subsequent trouble. Verbal instructions and suggestions are obviously of advantage to the incoming shift, but sometimes these are neglected or hurried over to the detriment of the service. The station log is usually the main reliance of men going on duty, in the absence of specific information from the outgoing force, but some plant managers have recognized the need of a definite statement of conditions from those leaving the work and have prepared forms for this purpose which are fulfilling a highly useful function.

The information included covers in a

representative case a day report and a night report signed by the watch engineer in each case, and the data set forth include such questions as—hour at which watch engineer assumed charge; anything reported by predecessor as being out of order; anything reported as wrong during previous shift, with any stops required; whether gage-glasses were tested; plentifulness of coal supply; disposal of ashes during shift; condition of water supply, engines, generators, pumps and stokers; violations of orders during shift; condition of piping and connections to boilers; whether boilers and tubes were blown off at the proper time.

A similar but much more abbreviated report is prepared by the engineer in charge of the night watch. The regular station log of hourly and other readings is not interfered with by the report of plant condition thus required, and the latter is beneficial in fixing the responsibility for the handling of the plant at stated hours, besides serving as a memorandum of emergency work which should not be allowed to run over into a later shift's period. Thus the incoming force is in position to meet the service conditions most promptly and efficiently.

Such records are of the greatest value as evidence of the care with which a plant is operated, and if the manager has occasion to use them in a court case, for example, the slight cost of preparing them becomes as nothing in proportion to their helpfulness.

Climbing the Ladder

Those with an ambition to succeed and reach the top of the ladder in their chosen work, should learn early to obey instructions. It is human nature to regard one's own ideas on how to manage or to do given things better than those of the one in authority, and when he orders a thing done a certain way the tendency is for the subordinate to want to substitute another way.

He fails to realize that probably his chief devoted some time thinking and planning this particular thing, before even mentioning it to him, and has looked at it from all angles. Very likely the subordinate's plan was rejected along with a dozen others, before the final one was adopted.

It is not the intent to discourage anyone from having ideas. They may be good ones and very valuable and acceptable if advanced at the proper time and in the proper way. But no matter how good the ideas may seem to their originator, he should not deliberately go contrary to his orders, even in the smallest detail.

As a rule it is a characteristic of the really successful that they do exactly as they are told, no matter whether their chief is a strawboss, a foreman, a superintendent, or a board of directors. In fact, the higher a man goes the more important it is that he obeys his orders to the very letter.

Contrary to the plaint (usually from the lazy) that the good places are all taken, there never was a greater demand for men who can carry out the plans of the "men higher up." Almost invariably the latter became what they are by first obeying the instructions of others. They are exemplifications of the old adage, "Before you can command you must learn to obey."

There is no line of work in which this is more true than in the power-plant field, nor are the opportunities to rise to the top more numerous or more rapid elsewhere.

The Average Owner

Conditions in the engine and boiler room of the small isolated plant are often not worth boasting about. In fact, the farther the visitor stays from the power plant, the greater peace of mind he will usually enjoy. The equipment is sometimes second-hand and very often in a dilapidated condition. Some of it may even be in a dangerous operating condition. Everything, including the starter-and-stopper, points to the fact that the power plant is considered only as a necessary evil and, instead of taking its place as a valuable adjunct to a prosperous business, it is only endured because it is impossible to run the business without it.

The owner of a small but prosperous and growing manufacturing plant in the Middle West recently informed his visitor that if his factory were located in a city where central-station power was available, he would not take a second thought before he would do away with his troublesome power plant altogether and purchase power from the central station. In the next breath he asked whether, in the opinion of his visitor, an open feed-water heater would have any effect in reducing the amount of scale formed in the boiler. It is apparent, therefore, that this man wished to be informed, but that his mind was so taken up with the cares of his business that he had little opportunity to study the needs of his power department.

Later an opportunity was given the visitor to meet the engineer and fireman. He was informed that this man's duties were to start and stop the engine, blow the whistle, put coal in the furnace, clean out the boiler of scale on Sunday, sweep up the shop, run the drill press when required and be general handy man. His wages were fifteen dollars per week.

This is not an exaggerated case. It is the picture of the average owner of the small plant and the average steam engineer. Both need awakening. The average owner is open to conviction if his engineer can tell him with the authority that comes from knowledge the advantages that will be gained from much needed improvements. For example, it would only be necessary to put the case properly to prove to the owner that his power would cost him practically nothing if the exhaust steam from the engine were used for heating the shop rather than being allowed to waste into the atmosphere. This owner needed only to be convinced that a feed-water heater would be a profitable investment for his money before installing one.

A general toning up of the power plant would react on the engineer and, instead of being a roustabout, his duties in the power department would be so important as to require all of his time and attention.

There are a thousand and more power plants in the country similar to this one. The average owner is so busy making his factory pay that he neglects the power department. If he would but stop to consider, he would realize that the expense for his power has its share of influence upon his profits and that a little better engineer at a higher salary may save many times his extra cost by increasing the plant economy.

It is two hundred years ago this year that the first successful reciprocating steam engine was erected by Newcomen and Savery. Where would the world have been today if this had never happened?

Readers with Something to Say

A letter good enough to print will be paid for. Ideas, not mere words, wanted

Repairs to a Balanced Piston

The balance piston, Fig. 1. was supposed to help carry the weight of the piston valve and valve-gear of the 38x54-in. high-pressure cylinder of a marine engine. The balance piston was found in the storeroom. evidently having been condemned by some former engineer. Indicator diagrams and a little calculation showed that the piston had too much area for the work required of it. The weight of the valve-gear is about 1350 lb. and

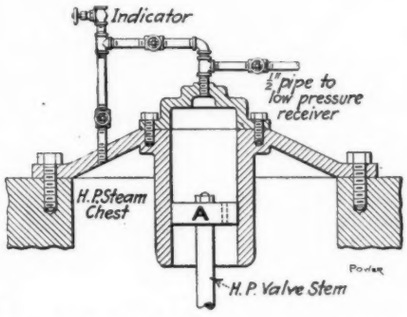


FIG. 1. BALANCE PISTON FOR CARRYING WEIGHT OF VALVE GEAR

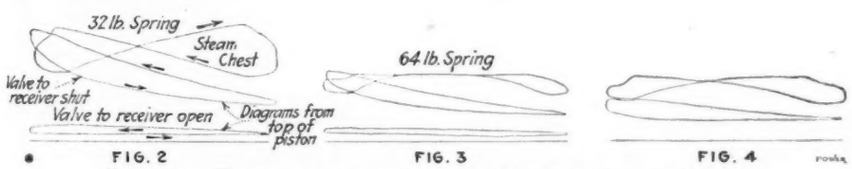
the diameter of the balance piston is 6 in. Fifty pounds per square inch would easily support this weight. but the boiler pressure of 100 lb. was applied. and as the top of the balance cylinder was connected by a pipe to the condenser, the pressure might increase beyond this amount. The pipes shown were connected to apply the indicator, and the pipe to the condenser was changed to connect to the low-pressure receiver, which carries a pressure of about 10 lb.

As the bottom of the balance cylinder is open to the main steam chest, the problem was to form enough back pressure on the top of the piston to reduce the mean effective pressure to the proper amount that would just balance the weight of the valve-gear. Diagram, Fig 2, was the first one taken, and it quickly explained the violent hammering which followed the attempt to use the whole power of the balance piston. There is considerable difference between the maximum and minimum pressures in every revolution, caused by fluctuations of pressure in the steam chest by the main throttle being partly closed. However. the noise was easily eliminated by closing an outlet to the receiver, but as will be seen from Fig. 2, the variation in pressure became greater by doing so, the steam on top of the piston being compressed on the up-stroke for lack of a free outlet. Should the valve to the receiver be closed altogether, this pressure would even exceed that in the steam chest, as shown by the shaded part of the diagram near the top center.

Fig. 3 shows that live steam was admitted to increase the back pressure. After some experimenting it was found that a ¼-in. hole *A*, Fig. 1, drilled through the piston supplied enough steam to maintain this back pressure while keeping the valve to the receiver open. Fig. 4 shows the final result. With this arrangement the piston has a mean lifting effect of about 1000 lb. without at any time taking the whole weight off the eccentric.

O. OLSEN.

New York City.



DIAGRAMS TAKEN BEFORE AND AFTER CORRECTING THE TROUBLE

Firing Boiler Furnaces

The following is a summary of my observations of methods of firing boilers worked at only a small fraction of their rated capacity. Boilers worked in this manner are very common in electric-light plants in small towns, especially plants supplying a day circuit. A few suggestions are also given.

In one town of less than 2000 inhabitants the load is so light for extended periods during the day that it scarcely shows on the ammeters. This plant is equipped with a first-class Corliss engine and two 66-in. by 16-ft. boilers, all in excellent condition. Only one boiler is in use at a time and it is always run with the ashpit doors and the damper wide open, regardless of the load that is being carried, the engineer claiming it will waste coal to partly close the dampers. In observing his method of firing, I found that the front part of the grates was well covered but the rear part was much neglected. Sometimes the interval between coal charges was so long that the fire became very thin, thereby allowing a large amount of cold air to pass through the grates. This is decidedly bad practice, but no trouble was experienced in keeping up steam due to the small amount of steam that happened to be demanded.

Whether the fire is thick or thin the grates should be well covered with fuel as any holes in the fire, and more especially those near the bridge-wall, are not conducive to good combustion or fuel economy. In the above case, with the light demand for steam, if the grates were kept properly covered with a fire of medium thickness and the dampers and ashpit doors were wide open it would be impossible to keep the boiler from blowing off, but the cooling effect of the air passing through the grates avoided this at the expense of the coal pile. Better results would be obtained by running a fire of medium thickness, paying particular attention to that part of the fire at the rear; the front part is usually taken care of pretty well, as most firemen are inclined to devote more attention to that part.

The fire should be given good draft for a few minutes immediately after firing, so as to give the fresh coal a good start and insure good combustion. The damper should then be partly closed; it will usually be found that quite a small opening at the damper will give very good results. This method will enable one to keep the grates well covered and also allow of a reasonable time between fires.

The previous paragraph may appear to indicate that I am not in favor of good draft. With a good strong draft and the proper amount of grate surface, the best fuel economy can be realized, but many boilers have a large excess of grate surface for the load that they are required to carry.

In a certain small town a new electric-light plant was started with a 100-hp. boiler supplying steam to a Corliss engine, but the peak load did not reach 50 hp.; most of the time the load was little more than the friction load of the engine. The rear half of the furnace grates was bricked over, resulting in a marked increase in fuel economy. This is good when practicable, but some similar plant may run along all day with little or nothing and have a peak load that will crowd

the boiler to its fullest capacity and then every square inch of grate surface is needed.

One capable engineer experimented with covering part of the grate surface with a slab of cast iron which was to be removed when the load came on. The experiment was not successful as difficulty was experienced in removing the slab because the slag and clinker from the coal had almost welded it to the grate bars.

One very good practice for such load conditions is to keep a very heavy bed of coal on the rear part of the grates, thereby effectively closing that part, and to do the firing on the front end. In this way the maximum draft can be used, and it has an advantage in that the fire at the rear can be broken, making the entire surface available at a moment's notice.

Another means for cutting down the grate surface, and still have it all available for peak loads, is to run a firebrick wall on the grates, lengthwise of the boiler, extending from between the fire-doors to the bridge-wall. This wall need not be over two or three bricks high and not more than one brick wide; a similar wall is constructed in the ashpit, only it should extend from the floor to the under side of the grates. By this arrangement the furnace will be divided into two distinct compartments. Only one side is used at a time as the other side is kept banked. It will be readily seen that with the ashpit door shut on the banked side there will be very little burning on that side, and in practice this plan has proved very satisfactory. When it is necessary to clean the fire or to use the entire furnace, the ashpit door on the banked side can be opened, the bank broken and in a short time the fire will be as hot as required. One advantage of this practice over that of keeping the rear end of the grates banked is that it admits of easily inspecting the fire; there is little excuse for permitting the banked fire to burn out. There may be some objection to having all the fire on one side during the light-load period but in practice this does not appear to cause any trouble.

It is so easy to maintain the necessary steam during these light-load periods, that the engineer is inclined to neglect to give the fire the proper attention. Much has been said and written about the detrimental effect of the infiltration of air through the cracks and crevices of the brickwork of a boiler setting. This is a good point and deserves attention, but more attention should be paid to the excessive amount of air that finds its way through the boiler flues by way of the holes in the fire.

The effect of excessive air passing through the fire was effectively illustrated to me during a recent visit to a large plant. This plant was equipped with steam flow meters on each boiler; at the load then being carried each meter indicated a steam flow of 10,000 lb. per hr. One of the boilers was not working properly and the flow dropped to about 2500 lb. These boilers were all equipped with chain grates, and investigation revealed that about 10 in. of grate surface was not getting coal, due to an obstruction in the coal hopper. This bare space extended the length of the grate, which was about 10 ft. As soon as the obstruction was removed the flow quickly came up to normal. This was no exceptional case and it would be difficult to tell how often it may have happened before, but with the indicating instruments it was at once apparent. In this case the turbines were getting all the steam they needed and no difficulty was experienced in maintaining the pressure. This condition may have continued for a number of hours and all that time this furnace was using nearly as much coal as the others with a very much smaller output.

G. S. SPRAGUE.

Geneva, Neb.

Stopping Leaky Cylinder Head Joints

A 20 and 38 by 42-in. compound engine developed a leak in the front head of the high-pressure cylinder, and as it meant much work and time to disconnect the parts and piping, loosen the foundation bolts, and move the cylinder back to put in a gasket, we decided to make a rather novel repair. The chief suggested that we remove the piston and put a strand or two of packing and some fine copper wire in the 0.025-space between the counterbore and the head and then caulk it tight. This was done and the engine was then put on the head end center and steam at 145 lb. pressure admitted; no leaks were noticeable.

The tool used for caulking was a 12-in. hacksaw blade with the teeth ground off, inserted in a plug and firmly secured by pins. This tool was driven by a small air hammer.

FRANK BUTTERWORTH.

Utica, N. Y.

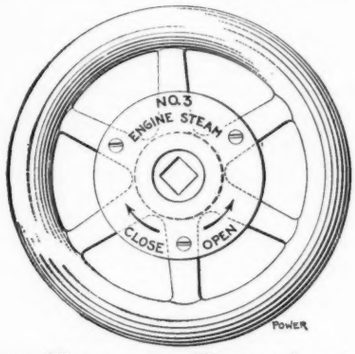
Indentification Markers for Valves

In large power or pumping plants where innumerable valves are used in connection with air, gas, steam, water and exhaust-steam lines, there is always chance, especially when there are many valves close together, that in a sudden emergency the wrong valve may be closed or opened.

This might lead to a serious accident, to both property and human life. To lessen this danger, throughout the plant I have charge of, I have installed safety markers on all the valve wheels.

The marker is in one piece, made of brass with the letters cast on. The letters have been buffed and polished while the background is painted dark blue to set off the letters.

Since some valves have rising stems while others have stationary stems, I designed the marker so that it would have a round hole in the center through which the stem could rise. The marker is securely fastened to the hub or spokes of



VALVE WHEEL WITH MARKER ATTACHED

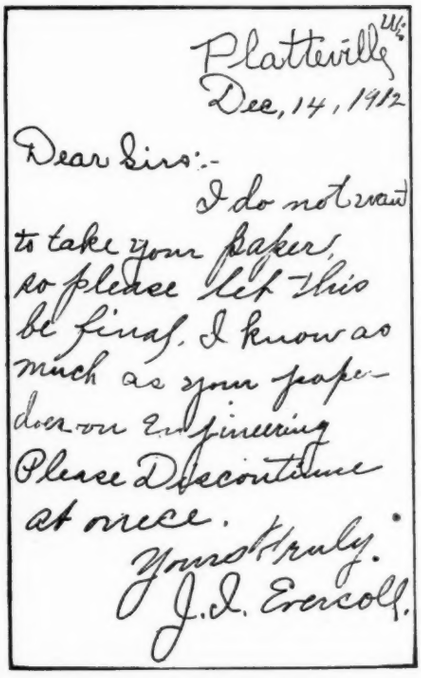
the wheel with countersunk machine screws.

In addition to the wording on the brass plate as to what the valve is for, I have also put on arrow marks, showing which way to open or close the valve. The sketch shows how this marker looks attached to a wheel.

A. RAUCH.

Swissvale, Penn.

A Reader with Something to Say


Platteville Wis.
Dec, 14, 1912

Dear Sirs:-

I do not want to take your paper, so please let this be final. I know as much as your paper does on engineering Please Discontinue at once.

Yours truly.
J. D. Ercoll.

WE ARE FLATTERED

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Cause of Explosion in Air Line

Referring to H. H. Delbert's trouble and inquiry in the Oct. 29 issue, I would suggest that, first, he use the best grade of air-compressor oil obtainable for lubricating the cylinders of his compressors, and, second, place an oil separator in the discharge line between the compressors and the air receivers, which will separate most of the oil that is being carried over into the receiver with his present arrangement. A separator similar to that used in ice-making and refrigerating plants for separating oil from the discharge, or high-pressure gas from ammonia compressors, would be very effective for this purpose.

The separator should have a gage-glass to show the amount of oil in it, and so indicate when it should be drawn off. Also, the separator should be placed as far from the compressors as possible to allow the vaporized oil time to condense so it will easily separate from the air. It is important that the separator should be large enough to allow the air to pass over the baffle-plates slowly to effect a more complete separation of the oil from the air.

I would also suggest that it is well to place drain elbows at all low points in the air-distributing mains and branches for drawing off water where necessary.

THOMAS BAKER.

York, Penn.

H. H. Delbert asks what caused the explosion in the discharge pipe of the air compressor and requests suggestions as to how to prevent a recurrence of this trouble. The receiver of an air compressor should be large, and placed as close to the compressor as possible, not over 20 ft. away. One chief function of an air receiver is to reduce the pulsations of the compressor, which are similar to water-hammer in a water line, and the greater the distance between the compressor and the receiver, the greater the shock due to the pulsations.

I was called upon to remedy a trouble in a certain plant, where the receiver was 300 ft. from the compressor, and the noise due to the pulsations of the air entering the receiver, sounded similar to striking the receiver with a sledge hammer. A number of fittings on this line had been broken. The trouble was eliminated by placing the receiver close to the compressor.

A second function of an air receiver is to cool off the air as soon as possible after it leaves the compressor and before it enters the pipe lines. In cooling off, the air gives up moisture, and the less moisture that enters the air lines the better. The air, in expanding through the different machines, cools in proportion to the expansion, and any moisture in the air freezes and chokes the exhaust ports. The only remedy then is to reheat the air as it enters the different machines, so that the larger the cooling system, the less water will go into the pipe lines.

Another precaution is to use a cylinder oil made purposely for the work. Poor oil has been the cause of a number of receiver explosions. If the compressors are properly designed and set up, and if Mr. Delbert will follow these instructions, he will have no further trouble with explosions in the discharge pipe.

DAVID BILLSON.

North Chelmsford, Mass.

Replying to Mr. Delbert's inquiry, I would suggest that one or more of the air-compressor valves leaked, causing the air to be compressed over and over until it became of such a high temperature that the carbon deposit ignited, and in burning heated the fitting until it could not hold the internal pressure and so let go. I would suggest that Mr. Delbert put a fusible plug that will stand 400 deg., and a thermometer, such as is used in economizers, in the discharge line. If the trouble occurs again, the fusible plug should prevent an explosion.

E. A. YOUNG.

Isabella, Tenn.

More About Siphon Condenser Troubles

Mr. Marier's letter on siphon-condenser troubles, in the Oct. 22 issue, reminds me of my experience with them.

Some years ago I took charge of a plant where there was a 20-in. siphon condenser, the water supply for which was taken from a canal leading to a large waterwheel. The injection water-supply pipe entered the canal directly in front of the waterwheel and back of a grating which was placed in the canal to catch the foreign matter in the water. This arrangement was the source of various troubles.

In the spring when the warm rains melted the snow and raised the water level in the river, much dead grass and other material would be brought down and lodge on the strainer, seriously interfering with the water supply to the condenser, making it necessary to close the bulkhead gates and draw down the canal through the waterwheel to clean the strainer.

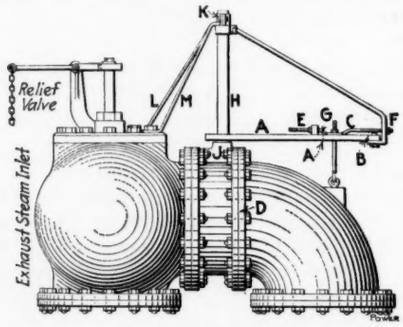
This sometimes had to be done twice a day, and it was frequently necessary to go down and cut the grass away with a stiff wire brush. At times in winter anchor ice formed in the river and in the morning we would suddenly be without water for the condenser, as the ice completely blocked the strainer, making it impossible to run until the ice was removed.

Whenever the river became low, the rush of water to the wheel churned air into the water, so that it became a serious menace, and if when the wheel was not running the water became too low, whirlpools would form at the end of the pipe and admit much air. In summer when the supply was insufficient to permit running the wheel, the water in the canal, on account of being of small volume, would get very warm, the temperature rising to as high as 80 or 85 deg. F. These troubles are not all peculiar to a siphon condenser, but are found with any type.

Previous to my taking charge of this plant there had been an accident to the water supply of the condenser just mentioned. A 2½-in. injector took its water supply from the pipe from the canal. I do not know whether there was a check valve on the line between the injector and the pipe at the time, but there was one after, and it was kept in good order. The steam blew back through the injector one night and somehow burst the 12-in. water pipe. The break occurred out in the yard where the earth contained many small gravel stones, some of which got into the pipe while repairs were being made, and I had the pleasure of removing them from the condenser cone and circulating pump.

Where I am at present there is a siphon condenser with a handy arrangement for lifting the elbow over the cone and swinging it out of the way for cleaning the cone. This arrangement may be easily and inexpensively made. It has

the advantage of being a part of the condenser and is always in place ready for use. A bar of flat iron *A*, 1x3½ in. in this case, has a slot 1 in. wide cut into the center from the points *B* to *C*, so that after the bolts are taken out of the flanges the elbow can be carried away from the joint *D* far enough to swing past the flange by manipulating the nuts



CRANE FOR SWINGING ELBOW TO ONE SIDE TO CLEAN CONDENSER CONE

E and *F* on the threaded ends of the carrying piece *G* and it may be raised by adjusting the nut on the hook in the eyebolt.

The mast *H*, which is of round iron, is pivoted in the block *J* and is free to turn in the collar at *K*. The braces *L* and *M* are attached to the studs on the flange as shown.

F. F. COWDEN.

New Bedford, Mass.

Economical Riddance of Turbine Scale

The article in the Nov. 19 issue, by Morgan C. Johns, on the above subject, has come to our attention. We experience trouble of the same kind as was referred to in our turbines and would like to have further details of the conditions under which Mr. Johns' turbines operate.

What is the steam pressure and superheat? What is the size of the turbine that can be cleaned with six barrels of coal oil? What type of boilers are in use? What is the analysis of the feed water? Does he have any trouble with his condenser from its heating up when the turbine is allowed to exhaust through it to the atmosphere? This question of scale in turbines is a very serious one with us.

R. J. C. WOOD.

Los Angeles, Calif.

In regard to scale in steam-turbine blades, I can say little, but I have had a 300-kw. turbine get badly choked with a fine substance that is sticky enough to close up the passages between the blades. There seemed to be no way to get this substance out, but to use small hooks and dig each blade free. We tried boiling the interior of the turbine with soda ash. We tried to soften the coating with oil, and then by applying a good strong air pressure to blow out the substance, but nothing had any effect. This trouble resulted from paper mill refuse being dumped into the water supply.

In summer, when the canal from which the water is taken is full, it does not harm, but in winter there is little water and much refuse, which causes the boilers to foam, and this fine dust seems to go with the steam. It lodges in the superheaters and traps and in any other place where there is a pocket in which it can lodge.

It is obvious, therefore, that there is danger of burning the superheater drum; even the tubes have to be watched, for they fill up with this stuff quickly. I do not understand how scale can form in the blades of a turbine unless there is some moisture in the steam, but even when a superheater is used there can be some water present at times. The turbine I read about in POWER could not have been coated with a very hard scale if it was softened and worked out so easily. I am watching for more letters about this scale question, for this turbine business is very interesting.

WALLACE JOHNSON.

Utica, N. Y.

Thermometer and Mercury Column Reading on High Vacuum Condensers

In the article under the above title in the Nov. 19 issue, I note on page 738 that the writer states that whenever possible a thermometer should be installed to give the temperature of the exhaust steam as this temperature can be used as a check on the reading of the mercury column.

I think that this point deserves a little further discussion because from the article one would be led to believe that the pressure shown on the mercury column should correspond to the exhaust steam temperature according to the regular tables of pressure and temperature relations of saturated steam. As a matter of fact the thermometer reading will always be that corresponding to some pressure lower than the mercury column reading. This difference is due to the fact that we always get some air mixed with the steam entering the condenser and the condenser pressure is the sum of the vapor pressures of both the air and steam present. The temperature reading for the exhaust, however, will be that corresponding to the vapor-pressure of the steam alone.

Thus in considering the temperature reading of the exhaust as a check for the vacuum reading, the check will be only to show the relative amounts of air and steam which are present in the exhaust.

The relation between the relative amounts of air and steam and their respective vapor pressure is one of direct proportionality of the relative volumes of the two to their separate vapor pressures.

JAMES L. HAYNES.

Milwaukee, Wis.

Split Pulleys

H. A. Jahnke's article on this subject in the Nov. 5 issue is interesting and full of facts. I had charge of the machinery and shafting in a factory for several years and found it hard to persuade the manager to stop getting solid pulleys. I remember once where we had to take off a 3-ft. diameter solid pulley. I could not drive it off or turn it on the shaft, so I put a chain on the rim and fastened the other end to a beam and put on the power slowly. I got the pulley to move along like a screw. I showed the manager next day the marks on the shaft and talked about split pulleys. I got them at last as we had so much changing of machines and pulleys that the advantage of split pulleys was apparent.

In the same factory I had quite a discussion with the foreman machinist. When the shafting was being installed the hangers were being put up with bolts through the floor beams. I proposed trying ¾-in. lagscrews. I made four screws as I could not get them quickly in that locality in those days—42 years ago. I tried the four screws on one of the hangers and a length of 2½-in. shaft, and had the boss machinist and the manager examine and test the screws. They were satisfied, so we had all the hangers put up with lagscrews, and made all the screws at the place—and there were 1600 of them.

Mr. Jahnke mentions split-loose pulleys. I had quite a number of them in an electric-light station in the early days of small units. The shaft was on the lower floor, with the belts coming up through the next floor to the machines. We were running the machines at different hours for incandescent and street lighting. Some of them had to be taken off the line at midnight. I did not wish to have the loose pulleys on the shaft, so I had a stand made with a loose pulley fitted on it. The loose pulleys were running only when the belt was thrown on or off.

In this same issue, Nov. 5, I noticed a picture of an Edison dynamo. We have three of them here in the factory which were made in 1884. These machines are still doing good work lighting the factory at 104 volts, which was the standard in the days when they were manufactured.

A. SANGSTER.

Sherbrooke, Que., Canada.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Effect of Lime on Pitting

When boiler-feed water contains a small quantity of lime, is it more or less likely to cause pitting of a boiler than when all traces of lime are removed from the feed water?

C. F. M.

Pitting is less likely to occur when the water contains a sufficient quantity of lime to form a thin internal scale of carbonate of lime.

Coils with Continuous Discharge

If coils are kept filled with steam at a given pressure, can water be heated by them as rapidly when the condensation from the coils is discharged by traps acting intermittently as when the discharge is continuous?

W. H. N.

No; a continuous discharge insures better removal of air and more complete contact of the steam with the interior surfaces of the coils.

Horsepower Constant

What is the horsepower constant of a 12x16-in. engine running 200 r.p.m., and what would be the horsepower developed with 19 lb. mean effective pressure?

R. S.

The area of the piston would be $12 \times 12 \times 0.7854 = 113.1$ sq.in. and the piston speed would be

$$\frac{16 \times 200 \times 2}{12} = 533.33 \textit{ ft. per min.}$$

The horsepower constant being the power that would be developed for 1 lb. m.e.p. is

$$\frac{1 \times 113.1 \times 533.33}{33{,}000} = 1.828$$

and with 19 lb. m.e.p. there would be developed $1.828 \times 19 = 34.73$ hp.

Advantage of Vacuum System of Steam Heating

What are the advantages of a vacuum steam-heating system over an ordinary gravity return system?

W. S.

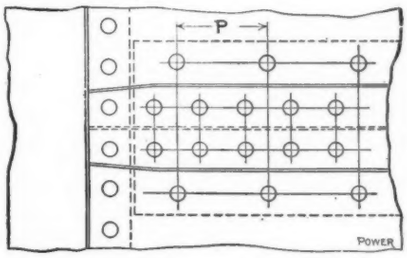
In a vacuum system, the discharge pressure being reduced, circulation and supply of steam to the heating surfaces are obtained by the use of lower initial pressure, which is especially desirable when the system is supplied with exhaust steam. A vacuum system also has better means for adjusting the activity of circulation and consequently the amount of heat radiated by the apparatus, as the discharging pressure and rate of air removal are both under control. It also possesses the advantage of requiring no air valves on coils or radiators.

Efficiency of Butt and Double Strap Joint, Double Riveted

What would be the efficiency of a butt-and-double-strap joint, double-riveted, with the tensile strength of the plate 55,000 lb. per sq.in.; thickness of the plate, $\frac{3}{8}$ in.; thickness of the butt straps, $\frac{5}{16}$ in.; pitch of the rivets on the row having the greatest pitch, $4\frac{3}{4}$ in.; diameter of the rivets after driving, $\frac{13}{16}$ in.; cross-section of the rivets after driving, 0.5185 sq.in.; strength of the steel rivets in single shear, 42,000 lb. per sq.in. of cross-sectional area, and in double shear, 78,000 lb. per sq.in. of cross-sectional area; and the crushing strength of the plates, 95,000 lb. per sq.in. of cross-sectional area?

E. S.

The figure shows a portion of a butt-and-double-strap joint double riveted in which the dimension *P* illustrates the unit



of pitch which is generally taken, viz.: the pitch of rivets on the row having the greatest pitch.

The efficiency would be the strength of the joint for a unit of length divided by the strength of the solid plate for a unit of length. The pitch being $4\frac{3}{4}$ in., if we take this as the unit of length then:

(A) The strength of the solid plate will be

$$4\tfrac{3}{4} \times \tfrac{3}{8} \times 55{,}000 = 97{,}969 \textit{ lb.}$$

The strength of the joint may depend on: (B) the strength of the plate between the rivet holes in the outer row, which will be

$$(4\tfrac{3}{4} - \tfrac{13}{16}) \times \tfrac{3}{8} \times 55{,}000 = 81{,}211 \textit{ lb.}$$

or (C) the shearing strength of two rivets in double shear, plus the shearing strength of one rivet in single shear, which will be

$$(2 \times 78{,}000 \times 0.5185) + (1 \times 42{,}000 \times 0.5185) = 102{,}663 \textit{ lb.}$$

or (D) the strength of the plate between the rivet holes in the second row, plus the shearing strength of one rivet in single shear in the outer row, which will be

$$[[4\tfrac{3}{4} - (2 \times \tfrac{13}{16})] \times \tfrac{3}{8} \times 55{,}000] + [1 \times 42{,}000 \times 0.5185] = 86{,}230 \textit{ lb.}$$

or (E) the strength of the plate between rivet holes in the second row, plus the crushing strength of the butt strap in front of one rivet in the outer row, which will be

$$[\,4\tfrac{3}{4} - (2 \times \tfrac{13}{16})] \times \tfrac{3}{8} \times 55{,}000\,] + [\tfrac{13}{16} \times \tfrac{5}{16} \times 95{,}000] = 88{,}574 \textit{ lb.}$$

or (F) the crushing strength of the plate in front of two rivets, plus the crushing strength of the butt strap in front of one rivet, which will be

$$(2 \times \tfrac{13}{16} \times \tfrac{3}{8} \times 95{,}000) + (1 \times \tfrac{13}{16} \times \tfrac{5}{16} \times 95{,}000) = 82{,}012 \textit{ lb.}$$

or (G) the crushing strength of the plate in front of two rivets, plus the shearing strength of one rivet in single shear, which will be

$$(2 \times \tfrac{13}{16} \times \tfrac{3}{8} \times 95{,}000 + (1 \times 0.5185 \times 42{,}000 = 79{,}668 \textit{ lb.}$$

For the assumed proportions and strength of materials it is seen from the above that the least strength of joint comes from consideration (G), i.e., 79,668 lb., then as the strength of the solid plate per the same unit of length would be (A), equal to 97,969 lb., the efficiency of the joint would be

$$\frac{(G)}{(A)} = \frac{79{,}668}{97{,}969} = 81\tfrac{3}{10} \textit{ per cent.}$$

Reducing Diameter of Eccentric

If $\frac{1}{4}$ in. is turned off an eccentric of an engine, what difference would it make to the valve travel?

H. A. G.

The throw of the eccentric and consequently the amount of the valve travel depend upon the amount of eccentricity, i.e., the distance from the center of the shaft to the center of the eccentric and the reduction of the diameter of the eccentric, as by turning off $\frac{1}{4}$ in., would therefore have no effect on the travel of the valve because it would not effect the eccentricity or throw of the eccentric.

Engineers' Study Course

Simple Lessons and Lectures for Home Study

Square Root

As we learned in our last lesson, a number is easily squared by simply multiplying it by itself. If the square is given, the process of finding the number which multiplied by itself will produce that number is not so simple, but it is not so difficult as it might at first appear. To find that number would be the process of finding the root and when we are dealing in squares it would be the process of finding the *square root* of that number.

Extracting square root is really nothing much more than long division and having learned long division, square root will come easy to you.

The first step in this process is to determine how many figures there will be in the root, and this is easily found by pointing off the given number into Periods of two figures each, commencing to point off at the right of the given number, i.e., the number the square root of which we are to extract.

We must always begin to point off at the decimal point.

The number of periods in the number indicate the number of figures there will be in the root.

Thus the square root of 65,536 must contain three figures for, dividing the number into periods of two figures each, there are (6'55'36) three periods. The last period contains only one figure, but it is nevertheless a period. The square root of 65,536 is

$$\sqrt{65,536} = 256$$

Notice the three figures in the root. Now to prove that the root 256 is correct, multiply

$$256 \times 256 = 65,536$$

When a number, the root of which is to be extracted, contains a common fraction, reduce the fraction to a decimal. If it is a decimal to begin with, then we do not need to reduce it, but in either case we start to point off each way from the decimal point. The number of periods in the integer and fractional parts of the number indicate the number of figures in the integer and fractional parts of the root respectively.

Thus the square root of 692.2161 is 26.31. Here there are (6'92.21'61) four periods in all, two in the integer part and two in the fractional part, so there must be two figures in the integer part and two in the fractional part of the root.

Example: If a square room contains 1156 sq.ft. of floor space, what is the length of one of its sides.

As the room is square the sides must be of equal length, and the area being their product, is the square of the sides. The length of either side, therefore, will be the square root of the area.

We first point off the number, 1156, into periods of two places each, commencing at the right hand. There will be two places or figures in the answer.

3	11'56) 34 root
3	9
60 trial divisor	256 dividend
4	256
64 complete divisor	

We find by trial the greatest number the square of which does not exceed 11, or the first left-hand period. This number is 3, the square of which is 9. We place 3 in the quotient, as in long division, and we also set it over to the extreme left, as shown. The quotient in square root examples is the root. Now we say 3×3 is 9 and place 9 under the 11, or the first left-hand period, and subtract, getting 2 as a remainder. Now we bring down the next period, which is 56, and place it alongside of the remainder 2. We now have 256 as a dividend.

We next place the 3 in the quotient under the 3 at the extreme left and add, getting 6, to which we add a cipher. This gives us 60 and is our trial divisor for the dividend 256. Now 60 will go into 256 four times, so place 4 in the quotient and add 4 to the *trial divisor* 60, which gives us a *complete divisor* of 64. Now we find 64 will go into 256 just four times. Hence 4 was the correct figure to be placed in the quotient and our answer is 34 as the *square root* of the floor area 1156 sq.ft. This 34 represents feet and is therefore the length of a side of the square room the area of which is 1156 sq.ft.

The square root of a number that is entirely a decimal will be wholly decimal.

Example: What is the square root of 0.000676?

Pointing off into periods we have

2	0.00'06'76) 0.026 ans.
2	4
40	276
6	276
46	

As the first period consists entirely of ciphers, the first figure of the root will be a cipher. Suppose the number, the root of which we are to extract, is not a perfect power, then the root will contain an indefinite number of decimal places.

But we can fix the number of decimal places by adding as many periods of two ciphers each as we wish decimal places in the root.

Example: Find the square root of 13 to three decimal places.

3	13.00'00'00') 3.605 ans.
3	9
60	400
6	396
66	40000
6	36025
7200	3975
5	
7205	

Here we added three periods of two ciphers each to the 13. The first figure 3 of the root is a whole number because its square 9 is less than 13. The rest of the process is readily understood.

As was said before, we begin to point off both ways from the decimal point in finding the square root of a mixed number.

Example: Find the square root of 324.7204.

1	3'24.72'04) 18.02 ans.
1	1
20	224
8	224
28	7204
8	7204
3600	
2	
3602	

Notice that we pointed off toward the left from the decimal point in dividing the whole number part into periods, and to the right from the decimal point when dividing the decimal part into periods.

When the last period of a decimal contains only one figure, add a cipher to complete the period.

In some examples in square root a peculiar condition arises, as shown in the following:

Example: Find the square root of 15,202,201.

3	15'20'22'01) 3899 ans.
3	9
60	620
8	544
68	7622
8	6921
760	70101
9	70101
769	
9	
7780	
9	
7789	

Referring, in this example, to the dividend 620, we find that the complete divisor 68 goes into this dividend 8 times, but subtracting $68 \times 8 = 544$ from 620 we have

$$620 - 544 = 76$$

for a remainder *which is greater than the divisor.*

But if we had said 60 as a trial divisor would go into 620, nine times, then we would have had

$$69 \times 9 = 621$$

which is 1 greater than the dividend and so cannot be used.

From this we learn that whenever the divisor, 68 in this case, goes into the dividend and leaves a remainder greater than itself when a divisor of one unit greater would more than equal the dividend, we must, nevertheless, use the lesser number, i.e., we must always have the product of the divisor and the figure in the root less than the dividend.

Example: What is the square root of 113?

1	113) 10.63 ans.
1	1
200	01300
6	1236
206	6400
6	6369
2120	31
3	
2123	

In this example the trial divisor 20 will not go into 13, so bring down a period, place a cipher in the root and also add a cipher to the 20, making it 200. The rest is easily understood.

Rules

(1) *Begin at the decimal point, or, what is the same thing, units place, and point off the number into periods of two places each.*

(2) *Find the greatest number the square of which does not exceed the first or left-hand period. This number will be the first figure of the root and should be written in the root and also at the extreme left of the given number. Multiply the number at the left by the figure in the root and subtract the product from the first period. Now bring down and annex the second period to the remainder.*

(3) *To the number in the column at the extreme left add the first figure of the root and annex a cipher to the sum. This figure will be the trial divisor.*

Divide the dividend by the trial divisor to obtain the second figure of the root. Now add the second figure in the root to the trial divisor to obtain the complete divisor. Multiply the second figure in the root by the complete divisor and subtract the product from the new dividend.

Bring down the third period and annex it to the remainder resulting from the subtraction. Now add the second figure in the root to the last divisor and annex a cipher to form a new trial divisor.

(4) *Work out the example according to the above rules until the last period, and then, if any more figures are required in the root annex periods of two ciphers each until the root contains the number of decimal places desired.*

(5) *Should the trial divisor not go into the dividend, place a cipher in the root, add a cipher to the trial divisor and bring down another period, then proceed as before.*

(6) *Should the root contain an indefinite number of decimal places and it is desired to have it contain a certain number of places, work the example to one place farther than desired, and if the last figure is 5 or greater, increase the last figure in the root by 1.*

Examples for Practice

(1) Find the square root of the following numbers: (a) 10.791, (b) 326.-1636, (c) 0.000784, (d) 15,202,201.

(2) (a) Find the square root of 7 to four decimal places, (b) of 13 to five decimal places.

(3) What is the length of wall required for a square engine room which must have 1764 sq.ft. area?

(4) To find the diameter of a circle that will inclose a given area, divide the area by 0.7854 and extract the square root of the quotient.

What would be the diameter of a cylindrical tank, the cross-section of which is 5674.515 sq.in.?

Answers to Last Week's Problems

(1) 20,449; 283.5856; $\frac{4}{9}$; $351\frac{9}{10}$.

(2) 2744; 599,077.1; $\frac{125}{729}$; $14{,}041\frac{65}{512}$.

(3) (a) That the square root is to be extracted. (b) it indicates what root is to be extracted.

(4) 576; 110,592; 104,976; $\frac{1}{512}$.

(5) That 43 is to be raised to the third power and its square root extracted. That 16 is to be raised to the fourth power and its square root extracted.

(6) 324 sq.in.; 315 sq.ft.

(7) 729 cu.in.

CORRECTION

A serious error appeared in the Engineers' Study Course for Dec. 17. The rule for ratio should have been expressed

$$Ratio = \frac{antecedent}{consequent}$$

The antecedent and not the consequent is the dividend.

No. 1 answer to last week's problems should be 12,325 instead of 12,609. No. 4 answer is for 18 per cent. less coal received; 21.32 tons for 18 per cent. more.

Study Course Exchange

The following readers are desirous of exchanging Study Course papers with others who are following the Course:

T. A. McGrath, New Alexandria, Penn., William H. Phillips, 218 East Genesee St., Buffalo, N. Y., J. P. Colton, Ohio City, Ohio; Henry Gronbach, Jr., 2409 Silver St., Brooklyn, N. Y.; Hans Frederiksen, Box 46, Nerrington, Conn.; William L. Keil, 1836 Etting St., Philadelphia, Penn.; J. G. Lee, Box 15, Goldfield, Nev.

Heating and Ventilation Course

A course in heating and ventilation will be commenced by the Harlem (New York City) Young Men's Christian Association on Jan. 9, 1913. The class is provided especially for engineers, draftsmen, mechanics and salesmen of heating and ventilating appliances, and will last 22 weeks. During the first part, gravity, steam and hot-water apparatus will be studied, including the proper proportioning of boilers, mains, radiators, chimneys and flues, and all special methods of steam and water circulation applicable to small buildings. The second part takes up methods of ventilation. This will include appliances adapted for office buildings, hotels, factories, etc., as well as such parts of the power plant as may be intrusted to the care of the heating engineer.

Some of the topics to be considered: Properties and nature of heat; loss of heat from buildings; heat given off from radiating surfaces; properties of steam; sizes of steam mains and return mains; design of heating system (high- and low-pressure, hot-water, vacuum and vapor systems); high- and low-pressure boilers; chimneys, breeching and grates; power-plant piping and equipment; ventilation of buildings (lofts and factories, stores and office buildings, schools and theaters); air treatments (filters and screens, air washers, humidity and humidifying apparatus).

Dalton's Law in the Steam Plant

By George F. Fenno

Air and steam pressure in the condenser and boiler and examples to show that a liquid at a given temperature continues to give off vapor until the pressure of its own vapor equals the maximum for that temperature.

In the realm of steam engineering there is probably no law more important than Dalton's. A knowledge of this law enables one to properly understand the behavior of mixed gases or vapors and the effect of the pressure of mixed gases on the formation of vapors. The engineer is principally concerned with the behavior of various mixtures of air and steam, or water vapor. A mixture of these gases is found in condensers, cooling towers, dry rooms and sometimes in boilers. Dalton's law may be stated as follows:

When two or more gases are mixed in a vessel the total pressure is equal to the sum of the pressures that each component gas would exert if it occupied the vessel alone, provided there are no temperature changes and there is no chemical action between the two gases.

In Fig. 1, if *A* contains 1 cu.ft. of dry air under an absolute pressure of 20 lb. and *B* contains 1 cu.ft. of water vapor at

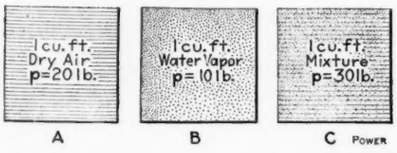


FIG. 1. PRESSURE OF COMBINED GASES

the same temperature as *A* and under an absolute pressure of 10 lb., then if these two gases are combined in a vessel *C* of 1 cu.ft. capacity, the total pressure will be 30 lb. Or, stating this in another way, the total pressure of 30 lb. in *C* is made up of 20 lb. air and 10 lb. steam pressure.

Of course, air itself is a mixture of different gases, thus the total pressure of the atmosphere is made up of the separate pressures exerted by the oxygen, nitrogen, carbon dioxide, etc., so that in reality in the above experiment, several gases were mixed and each one contributed its individual pressure to produce the total.

Every engineer knows that there is a definite relation between the pressure and temperature of water vapor. These relations are given in numerous steam tables, so that a ready method of determining the water-vapor pressures in any gaseous mixture containing steam is at hand. The curve in Fig. 2, giving these temperature and pressure relations of steam, is merely a graphical representation of the steam table, but it will show at a glance how to separate the partial pressure of air and steam. For convenience the pressures have been plotted in inches of mercury. Now suppose a condenser to have a vacuum of 27.5 in. of mercury (referred to a 30-in. barometer), then the absolute pressure in the condenser is 2.5 in. of mercury. If a thermometer were placed in any part of the condenser and showed a temperature of 80 deg. F., it would be known at once that the pressure the steam exerts at this place is 1 in. of mercury, as shown by the curve or a steam table, and the other 1.5 in. pressure must be due to some other gas in the condenser, presumably air.

That part of Dalton's law relating to evaporation, which perhaps is not so generally understood, may be stated as follows:

A liquid at a given temperature continues to give off vapor until the pressure of its own vapor equals the maximum for that temperature.

Vaporization, according to this law, will go on even though the space above the liquid is filled with some other gas at a higher pressure. Suppose when a boiler has been shut down for cleaning and is just being started up again, an air compressor is connected to the steam space and pumps up a pressure until the

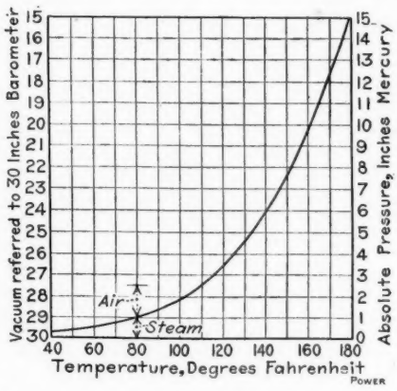


FIG. 2. TEMPERATURE-PRESSURE CURVE

gage showed 40 lb.; then suppose steam is raised until the gage shows 100 lb. What would be the temperature of the water in the boiler? Many engineers would say 337.9 deg. F., the temperature corresponding to 100 lb. gage pressure. But this is not the case, for there was 40 lb. gage, or 54.7 lb. absolute, air pressure to start with, so that, when 100 lb. showed on the gage, there would be only

$$100 - 54.7 = 45.3 \text{ lb.}$$

steam pressure present, and the temperature corresponding to this is 292.7 deg. F.

Nothing was said about the temperature of the air put into the boiler, but in order not to complicate the problem, suppose that at a temperature of 292.7 deg. it had a pressure of 40 lb. gage as assumed above. Of course, a change in the temperature of the air alters the pressure which it will exert.

If Dalton's law were not true, water could not evaporate into the atmosphere at a temperature of less than 212 deg. F., where its vapor pressure is 14.7 lb., or equal to that of the atmosphere. Yet all know that when clothes are hung on a line on a comparatively cold day they will dry, that is, the water evaporates. If the temperature of the air is 60 deg., the clothes will generate "steam" at a pressure of 0.5 in. of mercury. See Fig. 2. If this "steam" pressure of 0.5 in. were to remain pressing on the clothes they would never dry, as a greater vapor pressure could not be generated to push away the vapor already formed. Fortunately, air currents carry away the vapor and more "steam" is continually being generated, but always at a maximum pressure of 0.5 in. of mercury. If the day is very humid and "muggy," drying takes place slowly because the air already contains moisture exerting a pressure probably nearly up to the limit of 0.5 in. for the assumed temperature of 60 deg. F. If the pressure of the water vapor in the air itself before contact with the clothes were 0.5 in., the air would be saturated, that is, at that temperature it could carry no more water vapor. Strictly speaking, the air is not saturated, for, as Dalton's law tells us, the air itself makes no difference as to the amount of vapor or moisture present. In reality it is the space that the air occupies, that contains the moisture. However, for convenience, we usually speak of the amount of water that a cubic foot of air carries at a given temperature.

In a recent article showing how to calculate the quantity of heat necessary to carry on drying, the writer assumed that 966 B.t.u. were necessary to supply the latent heat of evaporation for each pound of water evaporated irrespective of the temperature at which the drying was done, 966* B.t.u. being the latent heat of vaporization at 212 deg. F. and atmospheric pressure. In the case just mentioned the temperature of drying happened to be 150 deg. and from what has just been said, it is evident that it is only the vapor pressure of the water itself which has any affect on the process of evaporation, so that the maximum vapor pressure on the substance being

*In accordance with the Marks and Davis steam tables the value 970.4 has now been generally adopted.

dried depended on the temperature. The steam table shows this pressure to be about 3.7 lb. abs. for a temperature of 150 deg., and the latent heat at this point is 1007.4 B.t.u., which is the figure that should have been used.

In a certain book on lubrication one finds the following: "The flash point is determined in the laboratory under atmospheric conditions. If the cylinder oil were to be tested under the high pressure carried in the steam pipe the flash point would undoubtedly be shown to be considerably higher, just as the boiling point of water, which at atmospheric pressure is 212 deg. F., increases with any pressure above that of the atmosphere (for instance, at 150 lb. per sq.in. the boiling point of water is 366 deg. F.). This will explain why it might be possible to use a cylinder oil successfully for lubrication under superheated-steam conditions where the temperature of the steam might even be a good deal higher than the flash point of the oil, measured under atmospheric conditions."

This author shares the quite common misunderstanding that the temperature of evaporation increases with any pressure. It only increases with the partial pressure of the vapor of the liquid itself.

Now the flash point, or the temperature at which ignitable vapors are given off, of most steam-engine cylinder oils is usually between 500 and 600 deg. F. They evidently then have to be heated to this temperature before their vapor pressure equals that of the atmosphere, so that at 366 deg. F. their vapor pressure must be quite low and the 150-lb. steam (a different gas) pressure can have no deterrent effect on the formation of the oil vapor. If the oil were tested in a closed room where the air pressure is 150 lb. the same flash point should result.

There are many other cases of the mixture of different vapors, such as in carburetting volatile liquids for internal combustion engines, which are interesting engineering examples of the application of Dalton's law. One more illustration, showing an unusual example, will suffice.

In cases of carbon monoxide poisoning, such as occur in mines, it is found that recovery is greatly quickened by placing the victim in oxygen at a pressure of two atmospheres. Due to the increase in pressure, the amount of oxygen in free solution in the blood, is increased immediately in accordance with Dalton's law and the tissues are more quickly supplied with the necessary oxygen. This oxygen is really forced into the blood and is quite independent of the normal exchange of oxygen that goes on.

Measuring Air from Compressor

By E. A. Fessenden*

Simple and accurate method of measuring air discharge from compressor. Only one tank is required and this may be the receiver forming part of the equipment.

*Associate professor of mechanical engineering, University of Missouri.

E. M. Ivens' article on "Air Compressor Efficiency" in the Oct. 15 number of POWER recalls to the writer a method of measuring the volume of air delivered by a compressor which he has often used successfully. While not any better than the orifice method or the method using two tanks suggested by Mr. Ivens, it may be of interest to readers.

Only one tank is required and this may be the receiver tank which often forms a part of the compressed-air equipment. If such a receiver is not a part of the equipment, any reasonably tight tank will answer the purpose if it is strong enough to hold the normal delivery pressure of the compressor safely. A few small leaks will not affect the results in any manner whatever.

The arrangement of the tank is shown in Fig. 1. The outlet pipe contains a tee for a thermometer cup *T* and two valves *A* and *B*. A carefully calibrated gage is provided to read the pressure of the air

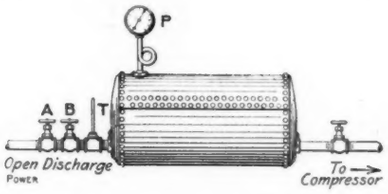


FIG. 1. AIR TANK FOR MEASURING DISCHARGE

in the tank. If the regular receiver is used the outlet pipe may be the ordinary service pipe, but when a test is to be made it must be disconnected from the service lines so as to discharge directly into the atmosphere through the two valves *A* and *B*. There should be a valve between the tank and the compressor. All other outlets from the tank should be closed off.

When ready for a test, first open valves *A* and *B* wide. Then with the compressor running at its usual speed gradually close valve *B* until the pressure in the tank remains constant at the normal working discharge pressure of the compressor. The compressor will then be running under normal operating conditions. After allowing the compressor to run in this condition for a short time to insure that all conditions have become constant, close valve *A* tight and allow the pressure in the tank to rise to 10 or 15 lb. above the normal working pressure. Then close the valve between the tank and the compressor and stop the latter.

Open the valve *A* quickly and take simultaneous readings of time and pressure as the pressure in the tank falls due to the escape of air through the valve *B*. The best way to do this is to have an assistant hold a watch and call "read" at uniform intervals of 15 or 30 sec. The exact pressure should be read at the instant the signal is called. Continue this until the pressure has dropped considerably below the working pressure.

Plot the readings just taken on cross-section paper so as to obtain a curve of pressure against time as illustrated in Fig. 2.

From the thermodynamics of gases we have the general equation for any gas,

$$PV = WRT \qquad (1)$$

in which

P = Absolute pressure in pounds per square foot;

V = Volume of gas under the pressure P, in cubic feet;

W = Weight of gas present in pounds;

T = Absolute temperature of the gas in deg.F. ;

R = Constant = 53.35 for air.

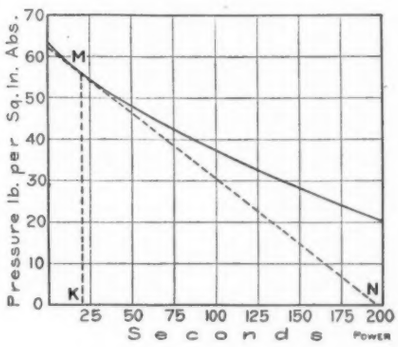


FIG. 2. PRESSURE-TIME CURVE

Equation (1) can be rewritten as

$$W = \frac{PV}{RT} \qquad (2)$$

In the case under consideration W, P, V and T represent the weight, pressure, volume and temperature of the air in the tank at any time. The volume and the temperature remain constant and the

pressure and weight decrease as the air escapes through the partly open valve B. The volume considered is the entire volume of the tank and all connections between valve B and the valve between the tank and compressor. This can be determined by filling the tank and connections with water, weighing the water required and calculating the volume by dividing by 62.355, the weight of 1 cu.ft. of water at 62 deg. F.

Differentiating equation (2) with respect to time gives

$$\frac{dW}{dt} = \frac{V}{RT} \times \frac{dP}{dt} \qquad (3)$$

in which $\frac{dW}{dt}$ is the rate of decrease of weight of air in the tank, or in other words, the rate of discharge, in pounds per second, while $\frac{dP}{dt}$ is the rate of decrease of pressure.

In Fig. 2 let M be a point on the pressure-time curve corresponding to the normal working pressure of the compressor. Draw the line MN tangent to the curve at M (this must be done carefully). Draw the line MK parallel to the pressure axis. Then

$$\frac{dP}{dt} = \frac{MK}{KN} \qquad (4)$$

The values of MK and KN are read in pounds per square foot and seconds directly from the figure.

If the value thus obtained for $\frac{dP}{dt}$ is introduced into equation (3) the rate of discharge in pounds per second is found directly. Since the weight discharged must be the same as the weight drawn into the compressor, the free air capacity is found by the formula

$$V_0 = \frac{WRT_0}{P_0} \qquad (5)$$

where V_0 is the volume in cubic feet of air per second at the intake pressure P_0 (in pounds per square foot absolute) and at the absolute temperature T_0, and W is the weight of air discharged per second just found from equation (3).

To illustrate this method consider Fig. 2 to be the pressure-time curve of a small compressor delivering air into a tank which with its connections has a volume of 47.7 cu.ft. The temperature of the air in the tank at the time the test was made was 87 deg. F. or 547 deg. abs. It was desired to find the amount of air discharged against a pressure of 56 lb. per sq.in. abs. Thus the line MK represents 56 lb. per sq.in. or 8064 lb. per sq.ft. From the figure KN is 194 — 20 = 174 sec. Then

$$\frac{dP}{dt} = \frac{MK}{KN} = \frac{8064}{174} \textit{ lb. per sq.ft. per sec.}$$

From equation (3)

$$\frac{dW}{dt} = \frac{47.7}{53.35 \times 547} \times \frac{8064}{174}$$

$$= 0.0758 \textit{ lb. per sec.}$$

At atmospheric pressure and 32 deg. F. the volume of air passing into the compressor is

$$V_0 = \frac{WRT_0}{P_0} = \frac{0.0758 \times 53.35 \times 492}{14.7 \times 144}$$

$$= 0.9391 \textit{ cu.ft. per sec.} = 56.346 \textit{ cu.ft. per min.}$$

When accurate results are required, it is well to repeat readings for the pressure-time curve several times and take the average value of $\frac{dP}{dt}$ thus obtained. It is, of course, important that the pressure gage and thermometer used be carefully calibrated, and that the temperature of the air remain the same when the readings for the pressure-time curve are taken as when the valve B is adjusted. After once adjusting valve B it must not be disturbed until the test is finished.

This same scheme can be used to provide a constant load on a compressor for an economy test. It is only necessary to adjust the valve B to give the discharge pressure desired and then run the compressor continuously at the proper speed, allowing it to discharge into the tank and the air escaping through valve B. As long as the pressure and temperature of the air in the tank are constant the rate of discharge is constant. The compressor governor should be adjusted to keep the speed constant and the pressure will remain constant as long as the valve B is unchanged. The temperature of the discharged air is practically dependent upon the amount and temperature of the jacket-cooling water. If these are kept constant the air temperature will not change appreciably. The compressor is then allowed to warm up thoroughly so that all these conditions become constant and the test is begun by adjusting valve B, thus fixing the rate of discharge as determined from a pressure-time curve. The load will now remain constant and the readings of steam pressure, steam weight, indicated horsepower, etc., are taken in the ordinary way.

The method outlined above has several advantages: It is rapid, simple, practically and theoretically correct and can be made very accurate if care is taken in making the readings and in plotting the pressure-time curve. It is flexible and can easily be adapted to widely different conditions of operation. All leakage and friction losses are taken into account. No expensive apparatus is required and often not even a change in piping. The manipulations and calculations are easy.

A Pacific Coast company, now erecting a large hydro-electric plant, will use a transmission voltage of about 150,000 volts on a line about 275 miles long, terminating at Los Angeles. Climatic conditions in California are such as to permit of voltage transmission as high as 250,000.

Notes on Oil Burning

By William Kavanagh

Whenever oil is burned in the liquid state or in conjunction with coal, smoke is sure to form especially when the furnaces are being heated up or steam is first being generated. When oil is burned in the gaseous form, smoke is less likely, but even then some will form if the men attending the furnaces are not accustomed to burning oil, or if the burners are poorly proportioned or designed.

In general, oil burners of the spraying type will produce smoke; the amount of air required for smokeless combustion of sprayed oil is usually so large as to greatly lessen its calorific value. Within the last few years the newspapers have been strenuously urging the use of oil on our battleships, claiming "the use of oil will prevent the detection of our warships on account of the absence of smoke, especially when the warships are below the horizon or hidden behind a projection of land." This argument is usually followed by the assertion that the speed of the ships will be greatly increased from its use.

Those not conversant with oil burning generally believe the newspapers' contentions and argue that the universal use of oil on warships is only a matter of time.

Two kinds of oil are in general use for steam generation. One known as "fuel oil," and the other as "crude oil." Fuel oil is a product of the oil refinery while crude oil is taken direct from the wells. The calorific values of the two oils do not vary much, but crude oil has the highest. The difference in favor of crude oil can be easily lost if it is poorly handled. It will take in round numbers 185 gal. of fuel oil to equal one ton of coal of good quality, allowing 7.5 lb. to the gallon and 20,000 B.t.u. per pound. One ton of good coal will contain 28,000,000 B.t.u. and dividing this by the heating value of one gallon of oil gives about 187 gal. as the result.

The rapidity with which smoke can be formed when oil is improperly burned is one reason why some means should be adopted for generating steam before oil is used in the furnace. Starting with every boiler cold and steam yet to be raised and with no way of atomizing the oil, smoke will roll off in great clouds, and continue to do so until sufficient steam is generated to atomize the oil.

Compressed air, if available, may be used as an initial means of smoke prevention, but neither air nor steam will stop smoke forming if the oil is improperly handled or fed to the oil burners.

Oil has its advantages and disadvantages as compared with coal. Among the first are quick steam generation, less space occupied, easier handling or moving from place to place and, when used on ships, a greater steaming radius for

the same bunker capacity. Its disadvantages are greater danger from fire and explosion, due to gas formation, and if it is not properly fed to the furnaces and the flame should happen to die out there will be no heat in the furnace. Then an explosion is likely, for after the flame dies out the oil or gas may appear again at the burner nozzles and, accumulating, explode when the burner is again lighted. Usually this results in blowing off the furnace doors, back connection doors, etc., and the boiler-room attendants are likely to be injured.

Another disadvantage of oil is the rapid formation and deposition of carbon on the heating surfaces of the boiler which retards heat transmission. Coal also will deposit ash or carbon on the boiler surfaces, but this deposition is slow when compared with that of carbon from improper oil burning. Ever since oil has been used for generating steam the promotors of some particular oil burner have advocated air as the atomizer arguing "when air is used it is burned," and in this way the calorific value of the oil is increased. Even were it true that air could be burned and that its use augments the heating value of the oil, special arrangements would have to be made to obtain the air at sufficient pressure to act as an atomizer. But air is noncombustible, and its use will not add any heat to the oil. Its supply is universal and it can be used when and wherever one pleases if one has the means to store and compress it.

Numerous attempts have been made to construct an oil burner that would produce its own air pressure, giving to the air sufficient velocity for it to act as an atomizer. Other oil burners have been constructed with a view to self-gas production. In this type of burner the flame issuing from the nozzles strikes the burner, thus raising its temperature and breaking up the liquid oil into finely divided particles easily gasified.

Oil has been burned in the gaseous form directly. This manner of oil burning is effected by pumping the oil into an oil boiler, the furnace of which is fed with coal or oil. The oil is thus gasified and the gas is then allowed to flow through the burner in the different furnaces until it issues and is ignited by a pilot flame or jet. The oil boiler is similar to a steam boiler and is handled after the same manner. The oil is pumped to the boiler and is carried at a certain height in the gage-glass. When a sufficient pressure is generated the gas is used as above mentioned.

One great objection to using oil in the gaseous state is its tendency to carbonize or choke up the burner nozzles and prevent satisfactory burning of the gas. All attempts to burn oil in the gaseous form have resulted in this carbonization and stopping of the openings in the nozzles, thus requiring constant attention on the part of the firemen to keep the slits or openings clear to permit the emission of the gas to the furnace.

Points in Buying Mechanical Stokers

By H. S. Knowlton

Many important clauses are incorporated in specifications and contracts, the purpose of which is to safeguard the purchaser.

A number of these points are brought to the reader's attention in this article.

Among the functions of the operating engineer none is more important than advising the plant owner in the purchase of new equipment. The more able an engineer is in selecting the best apparatus, the greater his value to his employer and the more likely will the right types of machinery go into the plant. Today the plant owner expects more of his engineer than doing routine operating work, and often studying machinery contracts, specifications and correspondence in connection with former purchases has well repaid the engineer.

Suppose the plant owner becomes convinced that it will pay to buy a mechanical stoker, and he asks his operating engineer's aid in drawing up the specifications and contract. Usually a consulting engineer will do all this work, but if the operating engineer is well informed and his judgment respected, he is likely to be consulted at least on the important engineering points which should be included in the specifications. The legal points to be covered are less his province, but even these are occasionally familiar to the operating man.

Legal Points

The legal phase of the stoker contract need not be discussd here in detail, but its general subjects and their significance are worth knowing. They include the date of the agreement, names of the parties concerned, definitions of any terms subject to false interpretation, description and quality of work required, date of completion, prices, payments and notice of shipment of the equipment as well as clauses covering the responsibility for accidents and damages incurred in connection with the work, method of securing alterations and amendments, guarantee of good workmanship and operating efficiency, abrogation of previous communications inconsistent with the terms of the agreement, and possibly a provision for arbitration in case of a dispute.

Sometimes there is a bonus or penalty clause for performing the work within a specified period, or based upon the efficiency of the equipment when tested under stated operating conditions. Ordinarily, the local engineer is not concerned with these points, but none of them should be overlooked.

Engineers' Problem

The engineering elements in a purchase are typified in a recent installation of stokers under three water-tube boilers, these involving general requirements, drawings, description of work, rating of boilers and operating results. The engineer selecting the apparatus required bidders to submit detailed drawings to scale. He gave each the boiler heating and superheating surface, operating steam pressure, superheat and the width of the furnace and its depth from the dead plate to the bridge-wall. The purchaser agreed to provide labor for handling the stokers from the railroad siding to the boiler room, but the maker was required to provide all labor for erecting, including the stoker parts and supports, driving shaft and air-controlling doors.

The purchaser was further to provide the premises and furnaces for the stokers, build the brickwork and furnish the air equipment to a stated point, besides connecting the driving engines with the line shaft. The engineer in charge defined the term normal boiler rating, giving the bidders the amount of evaporation per hour from and at 212 deg. and the superheat and steam pressure, based on 0.56 as the specific heat of superheated steam.

Operating Results

Probably the most important thing done by the engineer is what the owner particularly expects of him, to specify the operating results. In this case the engineer required the stokers to hold the steam pressure within 5 lb. of the desired working pressure when the boilers were normally operated at not over 100 per cent. in excess of their rated capacity.

Another important specification was that the stokers, when burning semi-bituminous coal of not less than 14,000 B.t.u. per pound of dry fuel, should give with the boilers and furnaces a combined efficiency of not less than 65 per

cent. when the boilers were operating at not less than 175 per cent. of their normal rating. The brickwork, however, was to be in good condition and free from cracks, and the baffling so arranged and the tube surfaces kept so clean that the temperature of the flue gas would not exceed 550 deg. It was also required that the stokers, when operated according to the maker's instructions and using the fuel specified, should burn 3300 lb. of coal per boiler per hour.

Conformation to the local smoke laws was required and the power for operating the three stokers and blower equipment was limited to 120 hp. when burning coal at the rate of 3300 lb. per boiler per hour. One fireman was to suffice for the three stokers when operated up to and including 200 per cent. of the boiler rating. The stoker manufacturer had to guarantee to replace, free of charge for one year, all repairs in excess of $350 or 5 per cent. of the stoker investment not caused by neglect or abuse. The dates and manner of delivery were also specified, the engineer recommending the purchase.

Obviously the detailed practice in purchasing stokers or other power-plant equipment would depend on the conditions; the figures cited were given merely as a concrete illustration. In another case, very different requirements might be proper, but in general the same ground should be covered by the operating engineer wishing to make a comprehensive recommendation when his employer asks his advice.

Reference in detail to a large amount of preliminary work has been omitted, since any recommendation would naturally be based upon a thorough study of competitive stoker designs, comparison of prices and guarantees, observation of the behavior of equipment in other plants and as wide reading as possible upon the general subject.

Care of High Vacuum Apparatus

Some practical suggestions on detecting and preventing leaks, the importance of using thermometers to show operating defects and the use of chemicals to detect contamination in water.

In plants carrying a high vacuum a large supply of cold water is absolutely essential. This may be curtailed by mechanical troubles in the pump or its prime mover, pipes too small, dirty strainers, pipes or tube sheets, and air leaks in the suction piping or at the shaft glands on the circulating pump.

Leakage at the glands is a most prolific source of trouble which can be overcome by proper ways of packing, such as the following which does not require frequent renewals.

Fill the stuffing-box for about one-third its depth with a soft metal packing, another third with a flanged metal ring and the rest of the box with the same soft metal packing. Then screw the gland up fairly tight. Drill down through the casing into the box near the inner flange of the metal ring and make a ¼-in. pipe connection to the discharge side of the pump. This forms an effective water seal that will fill any groove that may be cut in the packing and prevent air from flowing in.

The wet-air pump should be set with its suction inlet slightly below the condensed-water outlet in the condenser and a drain connection provided in the suction chamber, which should be opened when the air pump is stopped. Cast-iron flanged pipe and fiber gaskets are satisfactory on the air-pump line, which should be as straight as possible and as large as the pump outlet. The hotwell line should have a controlling valve near the pump and means for discarding contaminated condensate. If the piping be tight and the pump packing receives ordinary care, little trouble will arise. The water valves should be of composition to withstand the rise in temperature that will occur should the vacuum fail. Where there are more than one the air pumps should be interchangeable.

Dry-air pumps are generally of the rotative type with water-jacketed air cylinders and metal valves. If the piping is tight they give little or no trouble, the attendance required being only that needed by any reciprocating engine of the size and speed.

The hotwell pump, usually located about 4 ft. below the outlet of the condenser, is controlled by a float located in a chamber at the base of the condenser and adjusted to stop the pump while there is still a little water in the chamber. These pumps should be in duplicate to guard against failure.

CONDENSER LOG

Condenser No. 3 — Date July 13, 1911

Hour	Vacuum	Kw.	Temperature: Inlet water	Outlet water	Rear head	Steam inlet	Air pump discharge	Hotwell line	Remarks
6:30 p.m.	28.4	900	72	82	73	90	89	89	
7:00 p.m.	28.0	900	72	88	75	98	97	97	Strainer partially clogged.
7:30 p.m.	28.4	1160	72	82	73	90	89	89	
8:00 p.m.	27.8	1960	72	96	77	101	100	100	Insufficient circulating water.
8:30 p.m.	28.3	2260	72	83	73	92	91	91	
9:00 p.m.	28.4	2030	72	78	72	90	89	89	Too much circulating water.
9:30 p.m.	27.9	1860	72	82	73	101	100	100	Leak at glands plunger rods of air pump.
10:00 p.m.	28.4	1860	72	82	73	92	90	90	
10:30 p.m.	28.4	1800	72	81	73	92	91	91	
11:00 p.m.	28.4	1800	72	82	73	92	91	91	

The accompanying table is a condenser log, taken from a 2000-kw. Curtis horizontal turbine unit, covering a period of about 5 hr. At 6:30 p.m. the load was extremely light and the vacuum apparatus in normal condition. At 7 o'clock the vacuum had dropped to 28 in. without any change in the load. The increase in the temperature of the discharge water indicated an insufficient supply of circulating water. Investigation showed that the strainer on the intake was partially clogged. This was evidently cleaned, for at 7:30 the apparatus was in normal condition despite an increase in the load. At 8 o'clock the vacuum had fallen to 27.8 in. and the

Use of Thermometers

As a small change in the vacuum on a turbine makes a comparatively large difference in the economy, engineers have ceased to depend upon the spring vacuum gage as an indicator and have turned to the more accurate mercury column which in turn is checked by thermometers, for no change can occur in the degree of vacuum without a corresponding change in temperature. The connection of the mercury-column gage to the steam-inlet neck of the condenser is made as short as possible. Thermometer sockets are also placed in this neck and in the circulating-water inlet and outlet, the rear head of the condenser and the hotwell line. All thermometers should have a range at least up to 240 deg. F. Another thermometer is hung in the room adjacent to the mercury-column gage by which to correct the readings for temperature differences.

thermometer again showed a deficient supply of circulating water. This was corrected by throttling at the circulating-water inlet until the thermometer indicated the normal 10-deg. rise.

At 9:30 the vacuum had fallen to 27.9 in. The thermometers indicated that the circulating water was at normal temperature; hence the cause had to be an air leak. This was located at the glands on the air-pump plunger rods and corrected. The log shows normal conditions throughout the balance of the run. The table serves to show what valuable aid the thermometers render to the operator.

If centrifugal pumps are used to supply the circulating water, means are necessary for priming them quickly. A separate pump for this service, is generally provided, although any one of the wet-air pumps may be used, if means exist to discard the water that may be picked up in the operation.

Air Leaks

If the turbine-shaft glands are steam sealed, a gage should be mounted upon each packing case. The usual pressure maintained on these glands is 10 lb. gage and the steam just shows at the glands when the vacuum is normal. If any leaking occurs a greater pressure must be maintained to effectively seal the gland. If the glands are water sealed, there should be a water meter on the line supplying the sealing water or else an indicating valve should be used to control it. The normal amount of water for sealing purposes being known or the normal opening of the valve, a leak may be observed by the increased amount required.

Air leaks in the balance piping show by a marked rise in the degree of vacuum between no-load and full-load conditions. These leaks may or may not show when running noncondensing.

Leaks at the automatic atmospheric exhaust valve may be found by laying a temporary wooden cover over the end of the atmospheric exhaust, connecting a manometer gage to the pipe and working up a vacuum upon the unit. This may be done while the unit is in operation. Another method used where exhaust heads are placed on the end of the pipe, is to admit steam to the pipe on the atmospheric side of the valve and when the pipe is thoroughly heated, as shown by steam appearing at the end of the pipe, the steam is throttled until it just shows at this point. A vacuum is worked up and a leak at the valve will cause the small amount of steam to disappear entirely.

Leaks in the air-pump valves will show by the pump failing to maintain a vacuum within itself when the suction valve is closed. Air leaks in the condenser shell or in the piping may be found by the smoke test or by testing with a candle flame.

Testing Water for Contamination

When the circulating water is taken from a stream, be it either salt or fresh, there is always danger of contaminating the condensate through leaky tubes or tube packing. This may be guarded against by taking half-hourly chemical tests of the condensate. The necessary outfit consists of a dozen clear-glass 2-dm. phials, a dark blue or brown 2-dm. phial, a 4-oz. dark-brown bottle, a medicine dropper and a tray to hold them. If the circulating water is salt or brackish the chemical used is nitrate of silver, which is prepared as follows:

Weigh out 50 gm. of the silver nitrate crystals and place in the 4-oz. bottle, cover about 1 in. with distilled water (condensed steam), and shake until the crystals are entirely dissolved. Then fill the bottle to the top with distilled water. Next fill the dark-brown 2-dm. bottle two-thirds with the solution and put the medicine dropper in it. Cork up the balance and stand it aside until needed.

For a test take a 1½-dm. sample of the condensate from a convenient tap in the hotwell line by filling one of the clear-glass 2-dm. phials three-fourths full. With the medicine dropper held well above the mouth of the bottle containing the sample, press three drops of the solution into it. If a minute quantity of salt is present the sample will take on a bluish-white cast, but if much is present the sample will turn milky white. With a solution of this strength any water showing a trace of salt should be discarded until the leaky tube can be located and replaced. Care must be taken not to allow the dropper to touch the sample being tested, for if salt is present the small amount that adheres to the dropper will spoil the balance of the solution in the dark-brown 2-dm. bottle.

If the circulating water is fresh the method of procedure is the same, but the chemical to be used and the strength of the solution will depend upon the injurious ingredient in the water and the percentage that may be admitted to the system without giving trouble. The nitrate of silver test may be used if the water contains chlorides and litmus if acids or alkalies are present. Oxalate of ammonia will show organic matter.

Locating Leaky Tubes

Having ascertained that some tube leaks, the defect may be located in two ways. One is by flooding; that is, by closing the valve between the condenser and turbine and admitting water at about 10 lb. pressure to the steam space of the condenser after the water-chamber heads have been removed. The water finds its way through the leak and will be seen running out at the ends of the defective tubes. It is well while the heads are off to blow all the tubes with an air jet to remove any sediment that may have collected.

Often a tube will split and the flooding process fails to indicate the leak due to the pressure on the outside of the tube closing it sufficiently tight to prevent the water coming through. Then the condenser must be drained and the vacuum method applied by starting the air pump. The pressure of the atmosphere will force the split tube open and it can be located with a candle flame or by blowing a cloud of smoke against the tube sheet. Sometimes split tubes are difficult to locate and one end of the tubes may have to be blanketed by pasting wet newspapers over the tube sheet. This requires a little patience and skill to avoid tearing.

Before closing the condenser all the ferrules should be gone over to see that they are tight. It will save the trouble of taking off the heads at some future time only to find that the leak is at the packings.

Corliss Governor Troubles

By F. C. Holly

When putting a new governor in service or having trouble with an old one, knowing where trouble is most likely to be found, enables the engineer to quickly reason from the visible effect in the engine to the cause at the governor.

Probably the most frequent trouble in governors is binding in the governor-rod bearings, because the pins are too short; when the nut and washer are drawn up tight, side-binding results. This will cause a variation of engine speed in proportion to the friction induced, and a very little friction will cause a very noticeable variation at the switchboard; excessive binding will cause severe racing. This fault applies to the governor connections proper and to the cam-lever connections. The remedy is to adjust every rod so that it can be freely moved sideways on the pin.

In one instance, several years ago, an expert made a trip of 300 miles and back to loosen one little nut on one of the governor pins, this being all that was necessary to cure a bad regulating trouble. The engine was in a municipal lighting plant and at least one of the engineers was experienced in Corliss-engine operation.

The practice of placing the knockoff-cam lever bearing between the steam arm and the double arm, thus exposing it to a side thrust from each, is fundamentally wrong. The knockoff cam should operate on a separately flanged bearing.

Also important are the setscrews, or keys if any, in the governor gears. Serious accidents have occurred from these gears working loose on their shafts. On

one occasion an erector installed and started an engine and left it in charge of the regular attendant for a few moments. When he returned to the plant he found the engine racing badly. An investigation showed that the governor-driving gear was alternately slipping and catching on the shaft. The setscrew had not been properly tightened. Inspecting the governor gears and pulley occasionally is advisable.

Racing Causes

Sometimes, under extreme variations of load and even under less exacting conditions, a stiff governor will allow the engine to race. This is often due to too low speed in the governor, and is corrected by decreasing the size of the governor pulley and adding weight to the governor until the engine picks up its normal speed. The governor can be weighted by hanging small weights on temporary wire hooks to the projecting yoke of the governor, afterward weighing them and adding the equivalent weight permanently to the governor in whatever form its design best allows.

Before doing this, one should make sure there is no undue friction in the governor parts. This may be determined by removing the weights and balls and disconnecting the valve-gear, and seeing if the arms, after being raised as high as they will go, drop freely of their own accord. If, on moving them up and down, no binding or sticking is evident, there is no undue friction.

Dashpot Oil

A trouble every engineer experiences is that of oil getting low in the governor dashpot. This should be the first place to look for the cause of a racing engine; but much valuable time is often lost investigating other possible causes before looking to this most likely and obvious one. To make up for evaporation, a little oil should be added to the dashpot occasionally. When all governor parts are properly adjusted to eliminate friction and all bolts, setscrews and keys are tight, the possibility of trouble is narrowed down to slipping belts or the remote possibility of broken parts.

That all possible friction must be eliminated may seem unimportant but cannot be too strongly emphasized.

Counterweight Required

A leading pumping-engine builder contracted to furnish a high-duty pumping engine. The specifications required a governor capable of regulating the engine to a fixed speed. Previous to this contract no such regulating attachment had ever been furnished by this company, and none of the engineering force was familiar with governor design. However, a governor was made and applied. When the engine was started, and the governor put into commission, only the prompt action of the engineer in stopping the engine prevented a wreck. A more cautious trial was made and showed that the governor would lengthen the cutoff to the running away point and then go to the other extreme and shut the engine down.

Throughout the following months experiments were tried. Belt transmission was changed to gear drive and back again. Speeds were altered and cam attachments applied and taken off again; the best and oldest engineers of the company exercised their wits on the problem to no avail.

Extension rods ran from the governor to the valve-gear 25 ft. above, transmitting the motion of the governor arm through bell cranks and shafting. Hundreds of dollars were spent to say nothing of the loss of time and prestige before someone thought to counter-balance the weight of these rods and cranks and eliminate the friction from various pins and shaft bearings. Although these same pins and bearings were originally erected in a manner to satisfy all ordinary mechanical requirements, they were not perfect enough for governor work, a fact a good governor man could have pointed out in a few moments.

Sarco Automatic Lubricator

This lubricator is made in two types, one actuated by a pendulum, for moving bearings, such as crossheads, cranks, etc.; the other actuated by clockwork in a dustproof casing and used for stationary and slow moving bearings. Both lubricators feed grease similarly.

In the pendulum type, the pendulum, instead of a spring, actuates the clock movement and consequently when the bearing stops moving the pendulum the feeding of the grease stops simultaneously.

The body of the cup is of gun metal and screws to the bearing. Grease is pressed down into the bearing by a piston, operated by the clockwork, which is placed in a hermetically sealed casing. The speed of the piston depends on the the thread of the spindle, six different sizes being provided to suit conditions.

The clock is wound up daily. If more grease is delivered than required the clock stops automatically. It restarts as soon as the bearing requires lubrication. A lever on the cover of the lubricator releases and stops the clock. These lubricators are manufactured by the Sarco Engineering Co., 116 Broad St., New York City.

The Bureau of Mines has a copy of bulletin 49, "City Smoke Ordinances and Smoke Abatement," by S. B. Flagg, 55 pages, for distribution. Address the Director of the Bureau of Mines, Washington, D. C., and order by number.

Over the Spillway

Just Jests, Jabs, Joshes and Jumbles

Gold per fine ounce is valued at $20.671834625323. *Engineering News* asks why the computer "stopped at twelve places." For one thing, brother, this is cold weather. But twelve places! If we once stopped in this many, we would clamber up the H_2O vehicle and stick to celery tonic.

We take it, you have never yet tried to die a poor man. A. Carnaygie expects to do just this thing, leaving his heirs a paltry twenty-five million—and he is being severely criticized. When a man has a good impulse, encourage him; don't grab your 16-lb hammer and start the "Anvil Chorus." Even the millionaire is entitled to some consideration.

Somebody in sinking a shaft into the rich lodes of antiquity has discovered that old King Coal was a useful old soul even in Aristotle's halcyon days, about 238 B. C. Perhaps this is the reason we pay so much for it; it's a genuine antique. Johnny, run down into the cellar and chase up another hodful; mother is going to bake some biscuits.

A lady with the suggestive name of Catt urges woman suffragists to escape the thralldom of skirts and don trousers. They (the women) will break into the boiler room yet, and then you'll hear: "Why in ——! Beg pardon, Phyllis (bowing and opening the firedoor); when you have arranged your coiffure to your satisfaction, will you kindly fire up on No. 4? We're losing steam."

Who said corporations are soulless? The Hotel La Salle, Chicago, gave a Thanksgiving dinner to its 1260 employees. Because all the engineering Mahomets could not come to the mountain, a mountain of "big eats" came to them in the engine room. Gee whiz! Think of anybody discovering that there are human beings in an engine room!

In a South American town is a corking good machinist when he steers wide of the demon Rum. Red-eye and humor, mixed, is a sore combination in a power plant. One morning the master mechanic found the c. g. m. feeding a fine white powder to a compressor crankpin that was running hot.

"Shay, boss! Crankpin (hic) got a high fever. Givin' her quinine; 's only dope for fever!"

Economy in Steam Generation

By D. F. Nisbet

Calculation by formulas and approximate methods of determining the amount of air theoretically required to burn one pound of coal.

It is suggested that the volatile combustibles and fixed carbons be burned separately.

A recent trend is to increase boiler capacities far beyond their normal rating. While undoubtedly a move in the right direction, to be permanent, it must be carried out so that at least fair economy is obtained in operation. The furnace and its conditions must have more consideration than heretofore, and the proper burning of the fuel also will need to be treated as an engineering problem, and be studied as carefully and intelligently as has been the use of the steam. Putting a certain mount of coal into the furnace and an uncertain volume of air, leaving all other functional conditions to fortuity, will not be sufficient.

Most important is getting away from the need of admitting so much more air than the fuel theoretically requires, and, next, devising means for realizing more completely the heat value in the hydrocarbon distillates. Many other sins of omission and commission will need correction, but these two evils predominate and should receive first attention.

Many authorities on combustion maintain that 1 lb. of coal requires theoretically about 12 lb. of air, and, that practically 18 to 19 lb. gives the greatest efficiency. These statements have been widely disseminated, and would seem to assume that all bituminous coals require the same amount of air for proper combustion, and that 50 per cent. excess air is always necessary.

The weight of air theoretically necessary to completely burn a pound of any coal may be calculated from an ultimate analysis of the coal by the formula:

$$A = 11.52\,C + 34.56\left(H - \frac{O}{8}\right) + 4.32\,S$$

where A = air; C = carbon; H = hydrogen; O = oxygen; S = sulphur.

A close approximation to the heat value of the coals in British thermal units can be found also from the ultimate chemical analysis by Dulong's formula.

$$14{,}600\,C + 62{,}000\left(H - \frac{O}{8}\right) + 4000\,S$$

where C, O, H and S represent the percentage of the constituents of the coal.

To show that coals of variant chemical characteristics differ greatly in their theoretical air requirements, the ultimate analyses of four coals from widely separated localities are here given, each with the amount of air theoretically necessary for its perfect combustion and the thermal value.

Case 1. A Missouri coal, with an ultimate analysis of C = 52.55, H = 3.75, O = 7.10, S = 4.73. The air theoretically required is 7.22 lb. per pound of coal having a thermal value of 9465 B.t.u. Supplying 18 lb. of air per pound of this coal, the excess air would be 149 per cent.

Case 2. An Illinois coal analyzing C = 64.10, H = 4.54, O = 8.56, S = 4.87. The air theoretically required is 8.78 lb. and the heat value 11,705 B.t.u. With 18 lb. of air per pound of this coal the excess would be 105 per cent.

Case 3. A Hocking Valley coal having C = 68.3, H = 5.29, O = 15.64, S = 1.59. The air theoretically necessary is 9.09 lb. and the heat value 12,180 B.t.u. Per pound of coal this 18 lb. of air would show a 98 per cent. excess.

Case 4. A high-grade West Virginia coal analyzing C = 86.14, H = 4.41, O = 2.56, S = 0.60. The air theoretically required is 11.36 lb. and the heat value 15,135 B.t.u. With this coal an air supply of 18 lb. would be only 58.4 per cent. excess.

The four coals cited, show the fallacy of apportioning any arbitrary weight of air to all coals alike. That there is a well-defined relation between the thermal value of any given coal and the air required is also shown. This relation is very manifest when both the air supply and the thermal value are calculated from the ultimate analyses by the usual formulas, and where the heat units are derived by calorimetric determination, only a slight difference is observable.

Ultimate analyses, however, from which to determine the chemical minimum of air, are not always available, but the growing practice of buying coal on a heat unit basis has made quite easy procuring the proximate analyses and thermal values. When the heat value of any bituminous coal is known, a very close approximation of the theoretical air requirement can be found by multiplying this value in heat units by the factor 0.00075.

A more easily remembered rule is: every 1000 B.t.u. requires ¾ lb. of air. To find the air required then, simply point off three places from the right of the number of heat units and multiply by the decimal 0.75.

For instance: the coal cited in case 1 has a heat value of 9465 B.t.u. and

$$9.465 \times 0.75 = 7.098 \text{ lb.}$$

of air, which is 0.122 lb. less than 7.22 lb. found by calculatiing from the analysis.

The coal in case 2 has 11,705 B.t.u.,

$$11.705 \times 0.75 = 8.778 \text{ lb.}$$

which compares with 8.78 lb. found by calculation from the analysis.

Similarly in case 3 the value of the coal in B.t.u. is 12,180;

$$12.180 \times 0.75 = 9.135 \text{ lb.}$$

against 9.09 lb. by the usual formula.

In case 4 the value of the coal was 15,135 B.t.u. and

$$15.135 \times 0.75 = 11.351 \text{ lb.}$$

which is only 0.009 lb. less than the weight of air found by the formula.

Although only four coals are cited, they represent widely divergent analyses, and fairly cover the entire range of bituminous coals most used. The writer has applied this approximate method for estimating the theoretical air requirements to hundreds of analyses, and nearly always has obtained results agreeing with the amounts calculated from the ultimate analyses as closely as those here shown.

Since every excess pound of air passing through the furnace reduces the efficiency and these excesses in ordinary practice will range anywhere from 50 per cent. to 200 per cent. evidently, to secure economy in furnace operation, this is the first evil to be corrected. The use of such enormous surplusage of air is due to the construction of furnaces, both hand-fired and largely also, stoker-fired.

With present constructions, the air cannot be so evenly distributed that each combustible element will receive its quantity of oxygen and no more, nor can the intimate mixture be secured of air and gases, essential to complete combustion of the gaseous distillates from the coal. No matter how great affinity the gaseous combustibles have for oxygen, no chemical union can take place unless they are in intimate contact; certainly the greater the area of contact, or the more numerous the points of contact, the more rapid and complete the chemical union.

This inability to supply just the proper amount of air in just the right place, furnishes an excuse for using such excesses.

As a remedy, the writer can only suggest that in burning a fuel of dual characteristics like bituminous coal, its combustible elements being partly gaseous and partly solid, the volatile combustibles and the fixed carbons should be burned separately, with separate air supplies, each under control.

In such a furnace the element of chance could be largely minimized, the volatile combustibles more fully utilized, and excess air reduced, perhaps, to a negligible quantity.

Mixed Pressures in Boiler Plants

By T. M. Giles

Widely varying mixed pressures should not be allowed in plant operation unless the boilers on the low-pressure side can safely withstand the higher pressure for a limited time.

It is fairly common practice where conditions warrant, to operate a boiler plant at two different pressures. Generally reducing valves are used to maintain the pressures desired in the two parts of the piping system. Sometimes reducing valves are not used, but an ordinary stop valve is placed in the steam line to separate the high- and low-pressure parts of the pipe system, and this valve opened when it is desired to operate at one pressure. Can safety valves be arranged to adequately protect the low-pressure boilers in such a plant, when the pressures carried are considerably different?

The Massachusetts Board of Boiler Rules recognized the necessity of providing for the use of mixed pressure in boiler-plant operation, and has made rules governing the safety-valve equipment for such plants, but there seems to be very little reason for adopting some of the rules. They recognize three conditions of operation: First, where all the boilers are connected to a common steam main and are allowed the same pressure. In this case the pressure allowed and the grate area of each boiler determine the size of safety valve that is to be used for that boiler. This rule is perfectly proper and open to criticism in no way.

In the second place, the board has provided a rule to govern the equipment of a plant where the boilers are connected to a common steam main and are allowed different pressures, but are to be operated at one pressure. This rule provides that the safety-valve area on each boiler shall be at least that required for the grate area under it and the pressure that is allowed (not working pressure), and it is further stated that the aggregate valve area must be at least equal to that which would be required for the aggregate grate area, the lowest or operating pressure being used in calculating this valve area. The last part of this rule is intended to prevent the use of less valve area than would be required by the lower pressure in order to adequately relieve the pressure on all the boilers and prevent an excess of pressure on the weaker boilers. This rule would be logical if it also specified that the plant could not be operated unless all the boilers were connected to the system, but such a requirement would not be practical.

Apparently the effect of the discontinuance of one or more of the boilers on the rest of them was not considered. For example, under the present rules, if a plant containing ten boilers was allowed 160 lb. pressure, and equipped with 31.5 sq.ft. of grate area each, and two boilers allowed 95 lb. and with 32.5 sq.ft. of grate surface each, the safety-valve equipment would be as follows: The part of the rule referring to the minimum size of safety valve required for each boiler, would be met by using 3-in. diameter valves on the high-pressure boilers and 3.5-in. valves on the others; but nearly 29 sq.in. of total valve area would be lacking to comply with the latter part of the rule, referring to the aggregate valve area required.

This could be remedied by placing an additional 4-in. valve on each of the low-pressure boilers. If, for any reason, one of these low-pressure boilers was cut off the line, there would be a shortage of about 10 sq.in. of valve area for the remaining boilers, if they were considered as a complete plant under this rule.

Does the closing of the stop valve on a boiler or boilers supplying the excess valve area required under this rule, constitute a violation of it? This question may seem a little harsh and fine drawn, but safety valves have been condemned in Massachusetts on account of an excess of grate area beyond that specified of as little as ½ sq.ft., while under the above assumed conditions there would be a total excess of about 30 sq.ft. of grate area, or nearly 3 sq.ft. per boiler.

The third condition under which safety valves are installed where the boilers are connected to a common steam main, is where both the operating and allowed pressures are different. In this case, the lower-pressure boilers are to be additionally protected by safety valves placed on the pipes connecting them to the steam main, and such valves are to have an area at least equal to that of the connecting pipe. This rule seems to have been adopted on the assumption that a safety valve of a given diameter would be capable of discharging all the steam capable of passing through a pipe of the same diameter. This, of course, is by no means true, as may be shown by the Massachusetts rules for the determination of the size of safety valves, which are based on Napier's formula for the weight of steam discharged from an opening of a given area in a second of time at a fixed pressure. There is assumed an arbitrary rate of evaporation per square foot of grate surface to determine the ability of a boiler to generate steam. For pressures between 150 and 200 lb., the valve areas are based on the assumption that 200 lb. of steam might be generated per hour per square foot of grate surface, and since a valve of 4 in. diameter is considered capable of caring for 56 sq.ft. of grate surface under these conditions, it is estimated that it will successfully discharge about 11,200 lb. of steam per hour, or say 186 lb. per min. Tests of well designed safety valves show that this discharge rate is about what could be reasonably expected. At a pressure of 175 lb. gage, 186 lb. of steam per minute would represent a velocity of about 5000 ft. per min. when delivered through a 4-in. pipe. Such velocity in pipe lines is considered conservative for regular operating conditions, and can be greatly exceeded if a considerable drop in pressure is allowed. It will therefore be seen that it would be an impossibility for a 4-in. safety valve to discharge all the steam that could be transmitted through a 4-in. pipe.

To see how the safety of a plant complying with the Massachusetts law might be effected, assume a case of widely different pressures, where a small horizontal tubular boiler is used to furnish steam for heating, or other purposes requiring low pressure, in a manufacturing establishment. Assume that this boiler has seen its best days for other uses, and is now regarded as being safe for only 50 lb. pressure, which is ample for the work required. While this boiler is sufficiently large to care for the low-pressure work most of the time it frequently requires some assistance, and is therefore conneted to the high-pressure power boilers through a reducing valve. The regular boiler-plant equipment is composed of eight 500-hp. boilers, operating at 175 lb. pressure, so there is a difference of 125 lb. in the steam pressure carried on the two portions of the system. Assume that the low-pressure boiler is connected to the steam main by 20 ft. of 4-in. pipe which has placed on it a 4-in. safety valve, as required by the Massachusetts rule; also assume that the safety valve on the boiler is capable of caring for the steam generated by the boiler alone, and that both ends of the plant are working at rated capacity when the reducing valve becomes inoperative and allows the high-pressure steam to flow into the low-pressure side of the piping system. What chance is there under these conditions that the pressure on the heating boiler will not exceed 150 lb.? The eight 500-hp. boilers will contain about 384,000 lb. of water at a temperature of 377 deg. F., and assume that the pressure is to drop

15 lb. This would give a final temperature of 370 deg., or a drop of 7 deg. With the quantity of water given this drop would represent the liberation of about 2,680,000 B.t.u. or would produce the evaporation of 2250 lb. of water. A 4-in. diameter safety valve would require about 12 min. to discharge this amount of steam. A drop of 15 lb. from the initial entering pressure through the 20 ft. of 4-in. pipe connecting the low-pressure boiler with the steam main, would cause a flow through this pipe of about 770 lb. per min., or over four times the amount that the safety valve would be capable of discharging.

It should be remembered that the total flow of steam that has been considered is produced by stored energy in the heated water which is liberated by the reduction in pressure, and the furnace heat is left to supply the regular draft of steam to be used for operating the plant. It would seem that the explosion of the low-pressure boiler under the assumed conditions would be inevitable, unless it successfully withstood about three times the pressure that was considered safe for it. Apparently mixed pressures should not be allowed in plant operation, unless the boilers on the low-pressure side are considered safe to withstand the pressure allowed on the high-pressure side for a limited time. The fact that widely varying pressures are seldom used is perhaps one reason that the use of mixed pressures has not more often caused serious accidents in the boiler plant.

Procedure in Coal Specifications

By Mervin K. Baer

Pro rata corrections for heat values and deductions for excess volatile matter and ash. A convenient tabulation for comparison of bids.

That the great majority of consumers of fairly large quantities of coal purchase their fuel under specifications is well known, but how the various bids are tabulated so that they may be readily compared with each other is not generally understood. There are no hard-and-fast rules for writing specifications; they will vary with each individual consumer, according to the conditions which obtain. However, all such specifications take one of several general forms, the trend of which may be more easily followed by taking up a definite example under each class.

The board of education of a certain southern Michigan city recently bought coal under a contract which illustrated one type. The specifications call for "2000 tons of New River Sewell seam smokeless bituminous coal, mined in West Virginia, run-of-mine size (50 per cent. over pea size)." The standard upon which all computations are to be based is arbitrarily chosen in this case as follows: Heat value, 15,000 B.t.u.; volatile matter, including moisture, 22 per cent.; ash, 5 per cent.; these percentages being upon the basis of air-dried coal. No bids are to be accepted upon coals containing less than 14,250 B.t.u.; containing more than 25 per cent. volatile matter, including moisture; or containing enough sulphur to produce clinkers or give other unsatisfactory results. The correction factors to be used in arriving at the comparative value of the coal are given as follows:

1. A direct proportionality between the heat values.
2. Five per cent. of the price for each 1 per cent. difference in volatile matter, including moisture.
3. Two cents per ton for each 1 per cent. difference in ash content.

A set of specifications of this nature places the burden of the computations entirely upon the bidder; he must reduce his coal to the arbitrary standard set up by the specifications, and quote his price on this standard rather than upon his coal. Since all bids are on the basis of this standard coal, the purchaser can by inspection pick out the cheapest bid, or the bid he is willing to accept. It remains for him, however, to have an analysis of the fuel made upon its delivery, from which to figure the actual price to be paid, as seldom does the coal delivered have the same composition as the contract standard.

Suppose the analysis shows that the coal actually has the following composition: Volatile matter, including moisture, 23 per cent.; ash, 6 per cent.; heat value, 14,500 B.t.u.; and suppose the contract price to have been $3 per ton. The purchaser would arrive at the price he would have to pay in the following manner:

Correction for heat value:

$$\frac{14,500}{15,000} \times \$3 = \$2.90$$

Correction for volatile matter:

$$23 - 22 = 1 \textit{ per cent.}$$
$$\$2.90 \times 0.95 = \$2.755$$

Correction for ash:

$$6 - 5 = 1 \textit{ per cent.}$$
$$\$2.755 - 0.02 = \$2.735$$

the cost of the coal per ton.

Adopting a standard set of values is not very common at present. The more frequent practice is merely to set an upper limit for the various constituents, and have the bidders submit the prices and analyses of coals which will fall within these limits. The purchaser then tabulates the bids to have a comparative basis upon which to work.

This method is illustrated by the manner in which coal is purchased for a certain court house and post-office building. Bids are called for on 8500 long tons of Illinois Nos. 1 and 2 washed nut, whose upper limits in per cent. of dry coal are: Ash, 10 per cent.; volatile matter, 38 per cent.; sulphur, 2.5 per cent. The corrections in price are:

1. A *pro rata* correction for the heat value.
2. A 2c. per ton correction for each 1 per cent. difference in ash.

With this sort of a specification, the purchaser, upon opening the bids, constructs Table 1.

TABLE 1. TABULATION OF BIDS

Bid	(1) B.t.u. in coal as received	(2) Ash in dry coal per cent.	(3) Price per ton, Dollars Bid	(4) Price per ton, Dollars Plus ash difference	(5) Cost in cts. per million B.t.u.
A.....	11,400	10	3.02	3.100	12.140
B.....	14,800	6	4.10	4.100	12.367
C.....	11,350	10	2.80	2.880	11.328
D.....	12,136	8.9	3.04	3.098	11.396

The first three columns can be filled out directly, since these items are all specified in the bids. To compute column four, that coal which has the lowest ash content is taken as a basis, so that the corrections on all the other bids will be of the same sign. Thus, coal "B" is used as the standard; consequently it has no correction for ash. Coal "A," on the other hand, has an excess of ash over coal "B" of 4 per cent.; hence there must be added

$$4 \times 0.02 = \$0.08$$

per ton to the price of coal "A."

In column five, the corrections for heat values are made. Instead of using any one coal as a basis, however, the prices of all the coals are reduced to cost in cents per 1,000,000 B.t.u.:

$$\textit{Cost per } 1,000,000 \textit{ B.t.u.} = \frac{\textit{price per ton} \times 1,000,000}{\textit{B.t.u. per lb.} \times 2240}$$

For coal "A," this amounts to

$$\frac{310 \times 1,000,000}{11,400 \times 2240} = 12.14 \textit{ cents}$$

The last column reveals the comparative costs of the fuels: Coal "C" is the cheapest, at 11.328c. per 1,000,000 B.t.u.

Recently the method used in these two cases of correcting for ash differences has been much discussed, the consensus of opinion being that the price of the coal should not vary at a uniform rate throughout the range of its ash content, since coal containing 40 per cent. ash has no value for steam generation. In view of this, certain large consumers have adopted a general form for their ash specifications which is entirely different from those previously mentioned. The standard is a certain percentage of ash (depending upon the kind of coal required). For each 1 per cent. of ash less than this amount, a bonus of 2c. per ton is paid; for ash in excess of this standard percentage, the following penalties are deducted:

TABLE 2. PENALTIES FOR ASH

1 per cent. excess—	No penalty,
2 per cent. excess—	No penalty
3 per cent. excess—	2 cents per ton,
4 per cent. excess—	4 cents per ton
5 per cent. excess—	7 cents per ton
6 per cent. excess—	12 cents per ton,
7 per cent. excess—	18 cents per ton,
8 per cent. excess—	25 cents per ton,
9 per cent. excess—	35 cents per ton,

In addition to this ash correction, they use the *pro rata* correction for heat values.

It would be impracticable to cover all the different varieties of coal specifications now being used. The foregoing examples are given because they are thought illustrative of good practice in this rapidly grow[illegible] field of specifications for the purchase of coal, and because they also show that the work involved is not nearly as formidable as the title might imply.

Boiler Efficiency

The efficiency of a boiler is the ratio of the amount of heat which it absorbs and uses in making steam to the amount of heat furnished to it.

The heat which is furnished to the grate is all the heat contained in all the coal as fired.

But the heat furnished to the furnace and boiler is that in the coal which is *burned*. If a part of the good coal drops through the grate the boiler and furnace should not be charged with its heat.

The Committee on Tests of the American Society of Mechanical Engineers, therefore, recommends the consideration of two efficiencies.

The efficiency of the grate would equal the first of these formulas divided by the second or

$$\frac{\textit{Lb. combustible}\ \textbf{burned}}{\textit{Lb. combustible}\ \textbf{fired}}$$

Suppose a boiler burned 482 lb. of coal and evaporated 3856 lb. of water from feed of 130 deg. into dry-saturated steam of 150 lb. absolute pressure. The coal had a calorific value of 13,050 B.t.u. per lb. and 10 per cent. of ash, and there were 82 lb. of refuse taken out of the ashpit. What are the efficiencies of the combined boiler, furnace and grate; of the boiler and furnace; and of the grate alone?

To make one pound of steam of 150-lb. pressure from feed water of 32 deg. would take 1193.4 B.t.u. To raise a pound of water from 32 deg. to 130 deg., the temperature of the feed water requires 97.89 B.t.u. To make a pound of water already at 130 deg. into dry-saturated steam at 150 lb. would therefore require

$$1193.4 - 97.89 = 1095.51\ \textit{B.t.u.}$$

There are $3856 \div 482 = 8$ lb. of water evaporated per pound of coal, and since 1095.51 B.t.u. are absorbed for each pound of steam made, there are $1095.51 \times 8 = 8764.08$ B.t.u. absorbed for each pound of coal fired. The calorific value of 1 lb. of coal is 13,050 B.t.u. Then the efficiency of boiler furnace and grate is

$$\frac{\textit{Heat absorbed per lb. of coal fired}}{\textit{calorific value of 1 lb. of coal}} = \frac{8764.08}{13,050} = 67.16\ \textit{per cent.}$$

The portion not taken out of the ashpit was presumably combustible and burned. This is the difference between the amount of coal as fired and the amount of ashpit refuse, or

$$482 - 82 = 400\ \textit{lb.}$$

The water evaporated per lb. of combustible burned was therefore

$$3856 \div 400 = 9.64\ \textit{lb.}$$

And the heat absorbed per pound of combustible burned is

$$1095.51 \times 9.64 = 10,560.7+\ \textit{B.t.u.}$$

One pound of coal contains 13,050 B.t.u. Of this coal 10 per cent. or $\frac{1}{10}$ is ash. The other $\frac{9}{10}$ is combustible, and must contain all the heat units. If nine-tenths contain 13,050 B.t.u., one-tenth would contain $\frac{13,050}{9}$ and $\frac{10}{10}$ or a whole pound $\frac{13,050 \times 10}{9} = 14,500$ B.t.u.

$$\textit{Efficiency of boiler, furnace and grate} = \frac{\textit{heat absorbed per lb. of coal}\ \textbf{fired}}{\textit{calorific value of 1 lb. of coal}} = \frac{\textit{heat absorbed per lb. of combustible}\ \textbf{fired}}{\textit{calorific value of 1 lb. of combustible}}$$

$$\textit{Efficiency of boiler and furnace} = \frac{\textit{heat absorbed per lb. of combustible}\ \textbf{burned}}{\textit{calorific value of 1 lb. of combustible}}$$

Then the efficiency of the boiler and furnace exclusive of the grate is

$$\frac{\textit{Heat absorbed per lb. of combustible burned}}{\textit{calorific value of 1 lb. of combustible}} = \frac{10,650.7164}{14,500} = 73.45\ \textit{per cent.}$$

The number of pounds of combustible fired was $0.90 \times 482 = 4338$. The number of pounds of combustible burned was $482 - 82 = 400$. The efficiency of the grate was

$$\frac{400}{433.8} = 92.2\ \textit{per cent.}$$

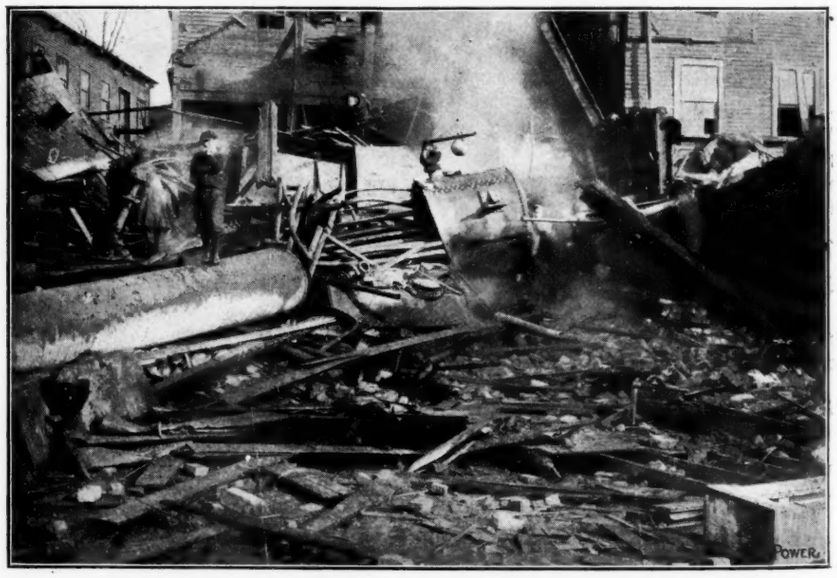
And the product of the efficiency of the boiler and furnace and that of the grate equals

$$\frac{10,650.7164}{14,500} \times \frac{400}{433.8} = 67.16\ \textit{per cent.}$$

as before determined.

Boiler Explosion at Keene, N. H.

The accompanying photograph shows the results of a boiler explosion on Dec. 5 at the plant of the Keene Glue Co. Fortunately there was no loss of life,



RESULTS OF BOILER EXPLOSION AT KEENE, N. H.

but considerable property damage resulted, as is evident in the photograph.

The boiler was a return-tubular, 16 ft. long, 66 in. in diameter and was made up of ⅜-in. plate. It was 20 years old and at the time of the explosion was carrying only 40 lb. pressure, its principal duty being to supply steam to the kettles and for other purposes in the factory.

The rupture of the boiler was caused through a lap crack which extended under the lap of the longitudinal seam for the full length of the middle course, then continued around the girth seams at the end of this course. The plate tore along the rivets for about one-third the circumference of the boiler and then tore through the solid plate diagonally from seam to seam. The shell plating was free from corrosion or pitting.

The boiler was insured and according to report had been inspected twice within the past nine months.

NEW PUBLICATIONS

HYGIENE FOR THE WORKER. By William H. Tolman and Adelaide Wood Guthrie. American Book Co., New York City. Cloth, 7½x5¼ in.; 231 pages; illustrated.

While this book is primarily designed for boys and girls from 13 to 18 years old, it is also adapted to the needs of all workers, old and young. Its purposes are twofold: "to equip the worker to care for himself under actual working conditions and to add to his happiness and efficiency."

The hygiene of the workroom, its proper conditions and ventilation, is carefully treated by the authors and many helpful suggestions are offered. Occupational dangers are also considered and recommendations made for the employee's protection. "First Aid to the Injured" is a valuable chapter, and "What the Worker Has a Right to Expect" makes clear the laws regarding hours of labor, what provisions are necessary for safeguarding the employee's rights and the sanitary regulations the state insists shall be carried out. There is also an interesting chapter on tuberculosis, how it develops and what precautions should be taken against contagion. The book is written in simple English and may be read with profit.

ANALYSIS OF METALLURGICAL AND ENGINEERING MATERIALS. By Henry Wysor, B. S. The Chemical Publishing Co., Easton, Penn. 10½x8 in., 82 pages, 9 illustrations, 2 tables. Price, $2.

The purpose of this manual is to furnish the student and chemist "the most practical methods compatible with accuracy and authoritative recognition for the analysis of such materials as are commonly brought to chemists in the field of metallurgical and engineering chemistry." Opposite each page of text is a blank page for the insertion of notes, equations, and the interpretation of results relative to the work to which the text refers.

Directions are given for the care of laboratory apparatus and the orderly arrangement of the chemicals. The book is devoted to practical direction for the sampling and analysis of fuels, gases, lubricating oils, ores of many kinds and the chief industrial metals.

The working chemist will find this book a valuable addition to his laboratory equipment.

The Elektrotechnischer Verein in Wien, the leading professional electrical society of Austria, announces the early appearance of a work which under the title "Statistik der Elektrizitätswerke in Österreich" will give a list of all the electric stations in Austria, Bosnia and Herzeogovina, with the names and addresses of the owners, the builders and furnishers of apparatus, number of inhabitants of the district served, and data regarding the operation, load, rates, etc. The work is published as one of the activities of the society and is sold practically at cost which is 4 kronen (eighty-odd cents) plus the postage. The address of the society is 12 Theobaldgasse, Vienna, Austria.

BOOKS RECEIVED

SOLDERING AND BRAZING. By James F. Hobart. D. Van Nostrand Co., New York. Cloth; 190 pages, 4½x7 in.; 62 illustrations. Price, $1.

PRIMARY BATTERY IGNITION. By C. Wadsworth, Jr. D. Van Nostrand Co., New York. Cloth; 78 pages, 4¾x7 in.; 26 illustrations. Price, 50c.

STEAM ECONOMY IN THE SUGAR FACTORY. By Karl Abraham; translated by E. J. Bayle. John Wiley & Sons, New York. Cloth; 5¼x7¾ in.; 104 pages; tables.

ALTERNATING-CURRENT MACHINERY. By William Esty. American School of Correspondence, Chicago, Ill. Cloth; 7¼x9¾ in.; 467 pages; 410 illustrations; tables and formulas.

SOCIETY NOTES

The semiannual meeting of the American Institute of Steam Boiler Inspectors, of Boston, began its season's educational work on Dec. 16, 1912. A dinner followed the meeting, the speakers being J. M. Winter, president American Institute of Steam Boiler Inspectors of New York City; John A. Stevens, chairman Massachusetts Board of Boiler Rules, and P. H. Hogan, past national president N. A. S. E. The organization, now in its third year, will continue to hold monthly meetings during the winter, at which papers and addresses of practical value to boiler inspectors will be presented. Social and educational advancement is the aim of the organization. Membership of boiler inspectors employed by the federal government, by state, cities and insurance companies is solicited. Communications should be addressed to E. R. Doherty, 31 Milk St., Boston, Mass., care of Mutual Boiler Insurance Co. of Boston.

Charleston Convention of Marine Engineers

The convention of the Marine Engineers' Beneficial Association will convene at Charleston, S. C., on Jan. 20, 1913, with headquarters at the Charleston Hotel. It is expected that about a hundred delegates will attend.

The local committee has its arrangements well in hand for the comfort and entertainment of the visiting engineers and their ladies, and promises to give them a hearty Southern welcome. The program will include a ball and reception, an oyster roast, a sail along the harbor and visits to near-by places of interest. A smoker under the auspices of the National Marine Engineers' Supply Men's Association, to be held in German Artillery Hall, on Wentworth St. will be a big feature, to which all prominent engineers and city officials will be invited.

The New York City delegation will leave for Charleston on Saturday, Jan. 18, by the Pennsylvania R.R., and every effort is being made to insure a full attendance.

OBITUARY

George B. Carpenter, founder and head of the railroad, mill and contractors' supply house which bears his name, died at his home in Park Ridge, Chicago, on Dec. 11. Mr. Carpenter was born in Conneaut, Ohio, in 1834 and moved to Chicago with his parents in 1850. In 1857 he acquired an interest in the concern of Gilbert Hubbard & Co. and in 1861, following the death of Mr. Hubbard, he obtained full control and reorganized the business under its present name. After the death of his wife in 1905 he retired from active business, although he retained the presidency of his company and continued as a director of the Great Lakes Supply Co. Mr. Carpenter is survived by four sons, Judge George A. Carpenter, and Benjamin, Hubbard F. and John A. Carpenter. The three last have been in active charge since their father's retirement.

PERSONAL

John J. Swan, formerly with the Chicago Pneumatic Tool Co., New York, as manager of the New York office and later as mechanical engineer in the compressor department, has become associated with the Oil Power Engineering Corporation, New York.

James F. Hobart, familiar to the mechanical reader as contributor to various technical journals, has recently resigned his position as inventor and designer in the research department of the Diamond Match Co., of Barberton, Ohio, to become superintendent of the Hanna-Breckenridge Co., Fort Wayne, Ind.

Moments with the Ad. Editor

Some men are superstitious enough to believe that '13 will, perforce, be an unlucky year.

It probably will be—for them.

We usually get things that are in harmony with our minds. If we conclude that a straight line and a wiggley one following 19 is going to send us to the Limbo of the unlucky, we'll get there no doubt.

For one mellow note from a flute there will be a dozen replies, some full and near, some vague and dream-like, but all in the identical key first ringing from the flute.

This principle, which is one of the most fundamental truths in music, is also one of the greatest and most valuable truths in life.

Since Nature controls the law of the echo business, you will get a tone in harmony with the one you send out.

If we are hopeful and optimistic, 1913 will bring us our share of its reward. If we persist in a belief that there is no hope in life and a bottomless pit for its end, we shall probably find no hope and a pit.

Following out the idea, if you strike a certain note in life—or in business—if the note is distinct, you are simply bound to get a response from every human being in key with that note.

In advertising, if you strike a note of sincerity, sincere people will listen to you. If the tone that runs through your business talk is genuine and honest, it is going to repeat itself in every genuine mind it reaches.

And the same rule holds when the tone sounded is false or tawdry or affected, for the first to respond will be the superficial and the veneered. They are the people invited, why shouldn't they come? People have realized this in a vague, instinctive way for ages but they haven't thought it out much.

The whole intention of modern business is to attract a favorable audience, to win the ear of the public, to establish good will between the world that buys and the world that sells.

Advertising is the main channel through which this is accomplished. No man can afford to float his business enterprize on any little side stream. And Advertising is the good, weather beaten old navigator who is willing and able to bring the bark to port.

Since advertising plays so important a part in the world's business the quality of it is a very important consideration.

In the first place, advertising was nothing more than a direct, honest foreword to the public. Then it was abused as every good thing is. The cheap and superficial, the insincere and the dishonest crept in. They sounded a morbid note—with its own answer—not a widely satisfactory one.

Folks found that a permanent business couldn't be built on anything but a fundamentally honest product.

And so the new school of advertising and publishing came into being—founded on the principle that there is no more potent argument for a good article than the truth about it and dedicated to the belief that the interests of subscribers must be considered above all things else.

1913 will see new things in advertising, greater progress, more intelligent copy, closer adhering to the square deal.

Incidentally, it will see the ad. editor located on page two, nearer to the beginning of things. We suspect our good friends will think enough of us to look us up at the new address every week. And you'll find us "at home."